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JOURNAL
OF THE
AMERICAN SOCIETY OF NAVAL ENGINEERS

VOL. XLIV. MAY 1932. No. 2

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ESTIMATED AND ACTUAL ENGINEERING PERFORMANCES U. S. AIRPLANE CARRIERS *LEXINGTON* AND *SARATOGA.**

BY LIEUTENANT H. L. DODSON, U. S. NAVY.†

The engineering data obtained during three sets of official trials conducted by the Board of Inspection and Survey for each of the aircraft carriers *Lexington* (*CV2*) and *Saratoga* (*CV3*) during the years 1928-30, together with the data obtained during precommission and shakedown periods, has been summarized and in some instances analyzed and is presented in the form of an easily accessible record in the following pages. Summaries of estimated per-

† Assisted by Lieutenants R. W. Bruner and C. J. Rend, U. S. Navy, Charles E. Ericson, Design Department, Fore River Plant, Bethlehem Shipbuilding Corporation, Quincy, Mass., Frank V. Smith, Marine Department, General Electric Company.

* The description of the machinery installations of the two vessels, together with certain engineering data, has been previously published in the following articles:

U. S. Aircraft Carriers, Mechanical Engineering, Volume 50, No. 4, April, 1928, pp. 280-284.

History, Description and Acceptance Trials of the U. S. S. *Lexington*, by Commander

formance prepared for these vessels are also included for the assistance that they will provide in the interpretation of the observed data and for their analytical value.

No attempt has been made to cover the performance of these vessels operating under service conditions with the fleet. The data presented are for steady conditions as exist on acceptance trials and on steaming runs in free route during shakedown periods.

The hull characteristics and machinery installations of both vessels are essentially the same; the machinery installations differing in boiler and main condenser design and in the design of certain auxiliaries. The boilers of the *Lexington* are of the Bethlehem-Yarrow type with Babcock and Wilcox superheaters; the boilers of the *Saratoga* are of the Babcock and Wilcox White-Forster type with Babcock and Wilcox superheaters. The main condensers of the *Lexington* are of the Navy standard type whereas those of the *Saratoga* are of the Lovekin type.

For a better understanding of the data in this article the following diagrams are included:

Figure 1—Propulsion Machinery Lay-out.

Figure 2—Selection Diagram for Main Turbo-generators and Main Motors.

Figure 3—Diagrammatic Arrangement of Steam Distribution Circuits for Normal Operation.

METHODS OF MEASUREMENT.

Standardization runs for both vessels were conducted over the Point Vincente, California, measured mile course in the manner customary for U. S. Naval vessels.

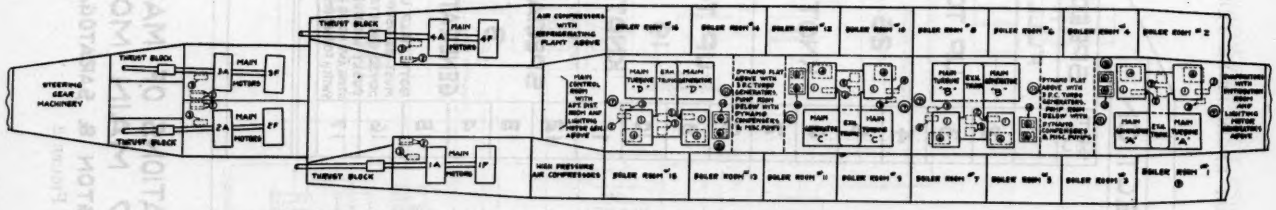
C. S. Gillette, U. S. N., JOURNAL A. S. N. E., Volume XL, No. 3, August, 1928, pp. 488-495.

The Propulsion Circuits of the U. S. S. *Lexington*, by Lieutenant E. G. Kranzfelder, U. S. N., JOURNAL A. S. N. E., Volume XL, No. 4, November, 1928, pp. 546-625.

Some Observations on the Design of Airplane Carriers and Notes on the *Saratoga* and *Lexington* by Rear Admiral George H. Rock, U. S. N., Transactions of the Society of Naval Architects and Marine Engineers, Volume XXXVI, 1928, pp. 63-76.

Trial Trip Data of the U. S. S. *Saratoga*, by Lieutenant Commander J. F. Alexander, U. S. N., Transactions of the Society of Naval Architects and Marine Engineers, Volume XXXVII, 1929, pp. 201-220.

Naval Electrical Power in Commercial Use—U. S. S. *Lexington* Relieves Tacoma Power Shortage, by Lieutenant Commander H. L. White, U. S. N., JOURNAL A. S. N. E., Volume XLII, No. 2, May, 1930, pp. 311-315.

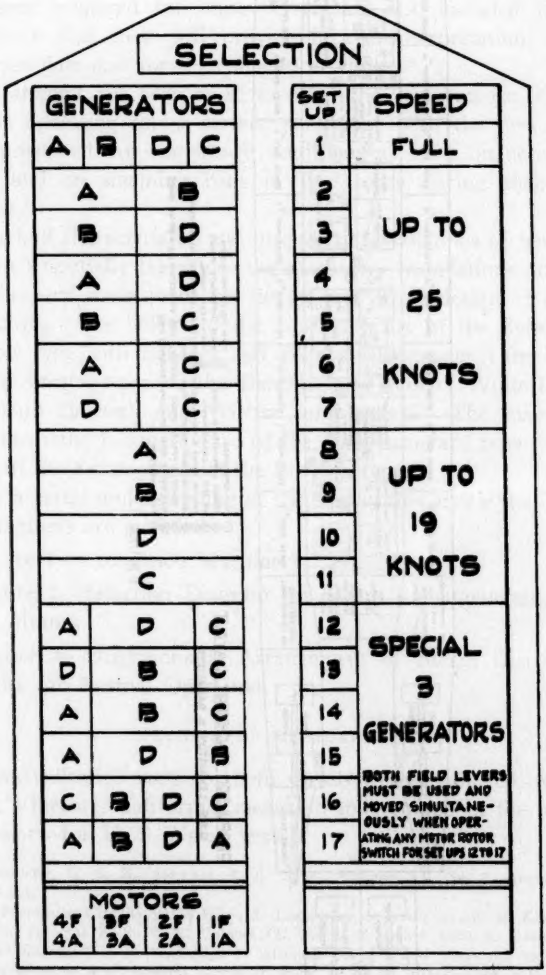

STEERING GEAR MACHINERY
THRUST BLOCK
MAIN MOTORS
AIR COMPRESSORS WITH REFRIGERATING PLANT ABOVE
MAIN CONTROL ROOM WITH AFT DIST. BRD. AND LIGHTING PANEL ABOVE
HIGH PRESSURE AIR COMPRESSORS
BOILER ROOM "6"
BOILER ROOM "5"
BOILER ROOM "4"
BOILER ROOM "3"
BOILER ROOM "2"
BOILER ROOM "1"
MAIN TURBINE "D"
EXH. TURBINE
MAIN TURBINE "C"
MAIN TURBINE "B"
MAIN TURBINE "A"
DYNAMO FLAT ABOVE WITH 3 D.C. TURBO GENERATORS, PUMP ROOM BELOW WITH DYNAMO CONVERTERS & MISC. PUMPS
CONNECTING WITH INTERCOM. BRDS. AND LIGHTING PANELS ABOVE

PROPULSION MACHINERY LAYOUT

U.S.S. LEXINGTON & SARATOGA

1. MAIN CONDENSERS
2. FORCED LUBRICATION OIL PUMPS
3. AIR AND OIL COOLER PUMPS
4. HOISTING TANKS & MAIN GENERATOR AIR COOLER HYDRAULIC VACUUM PUMPS
5. AUXILIARY AND RESERVE FEED PUMPS
6. 1 BOILER, 2 FORCED DRAFT BLOWERS, 1 FUEL OIL HEATER (EACH BOILER ROOM)
7. FEED TANKS, MAIN AIR PUMPS (RAPHEY) OVER
8. FEED HEATERS
9. MAIN CIRCULATING PUMPS
10. MAIN CONDENSATE PUMPS
11. MAIN FEED PUMPS

FIGURE 1.



POSSIBLE COMBINATIONS OF MAIN TURBO-GENERATORS AND MAIN MOTORS

U.S.S. LEXINGTON & SARATOGA

FIGURE 2.

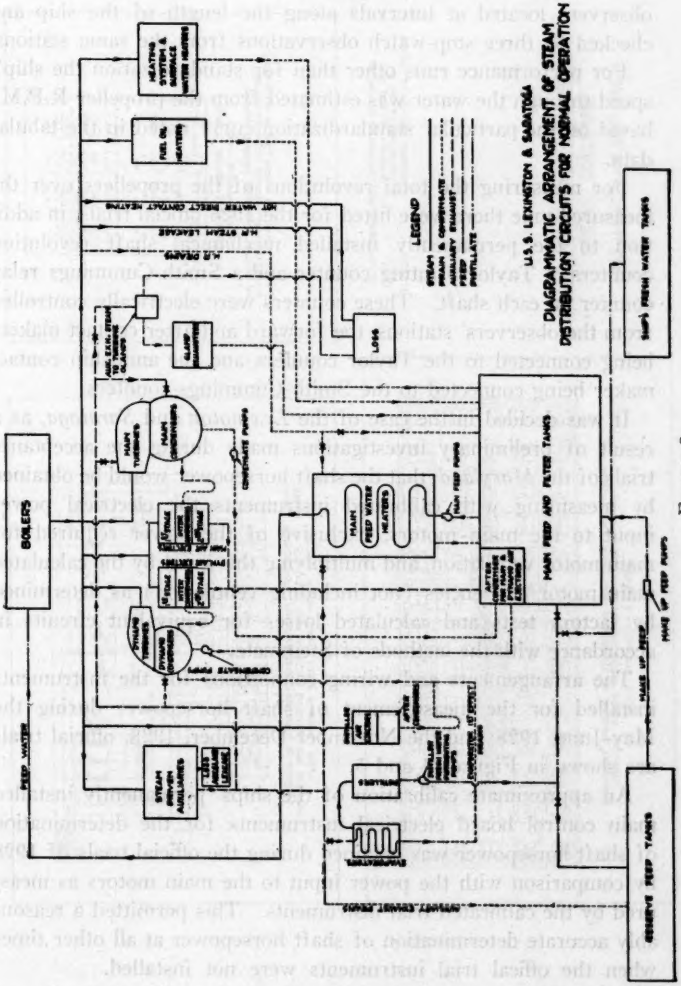

U.S.S. LEXINGTON & SARATOGA
DIAGRAMMATIC ARRANGEMENT OF STEAM DISTRIBUTION CIRCUITS FOR NORMAL OPERATION
LEGEND
STEAM
DRAIN & CONDENSATE
AUXILIARY EXHAUST
FEED WATER
PARTICULATE
BOILERS
MAIN TURBINES
MAIN CONDENSERS
MAIN FEED WATER HEATERS
MAIN FEED AND FILTER TANKS
RESERVE FEED TANKS
SHIP'S FRESH WATER TANKS
LOSS
MAIN FEED PUMPS
CONDENSATE PUMPS
MAKE UP FEED PUMPS
MAKE UP FEED
FEED WATER
STEAM DRIVEN AUXILIARIES
HEATING SYSTEM & SURFACE WATER HEATERS
FUEL OIL HEATERS
M.N. STEAM LEAKAGE
M.N. DRAINS
HOT WATER HEAT CONTACT HEATING

FIGURE 3.

Timing during standardization was obtained by the use of a chronograph operated by three contact makers in the hands of observers located at intervals along the length of the ship and checked by three stop-watch observations from the same stations.

For performance runs other than for standardization the ship's speed through the water was estimated from the propeller R.P.M., based on the particular standardization curve noted in the tabular data.

For measuring the total revolutions of the propellers over the measured mile there were fitted for the 1928 official trials, in addition to the permanently installed mechanical shaft revolution counters, a Taylor printing counter and a Smith-Cummings relay counter on each shaft. These counters were electrically controlled from the observers' stations, the forward and after contact makers being connected to the Taylor counters and the amidship contact maker being connected to the Smith-Cummings counters.

It was decided in the case of the *Lexington* and *Saratoga,* as a result of preliminary investigations made during the acceptance trials of the *Maryland,* that the shaft horsepower would be obtained by measuring with calibrated instruments the electrical power input to the main motors, exclusive of the power required for main motor ventilation, and multiplying this input by the calculated main motor efficiencies (not including ventilation) as determined by factory tests and calculated losses for equivalent circuits in accordance with the methods of Steinmetz.

The arrangements and wiring connections for the instruments installed for the measurement of shaft horsepower during the May-June, 1928, and the November-December, 1928, official trials are shown in Figures 4 and 5.

An approximate calibration of the ships' permanently installed main control board electrical instruments for the determination of shaft horsepower was obtained during the official trials of 1928 by comparison with the power input to the main motors as measured by the calibrated trial instruments. This permitted a reasonably accurate determination of shaft horsepower at all other times when the offical trial instruments were not installed.

The steam consumption of only one main turbine of each vessel was measured during the 1928 official water rate and fuel con-

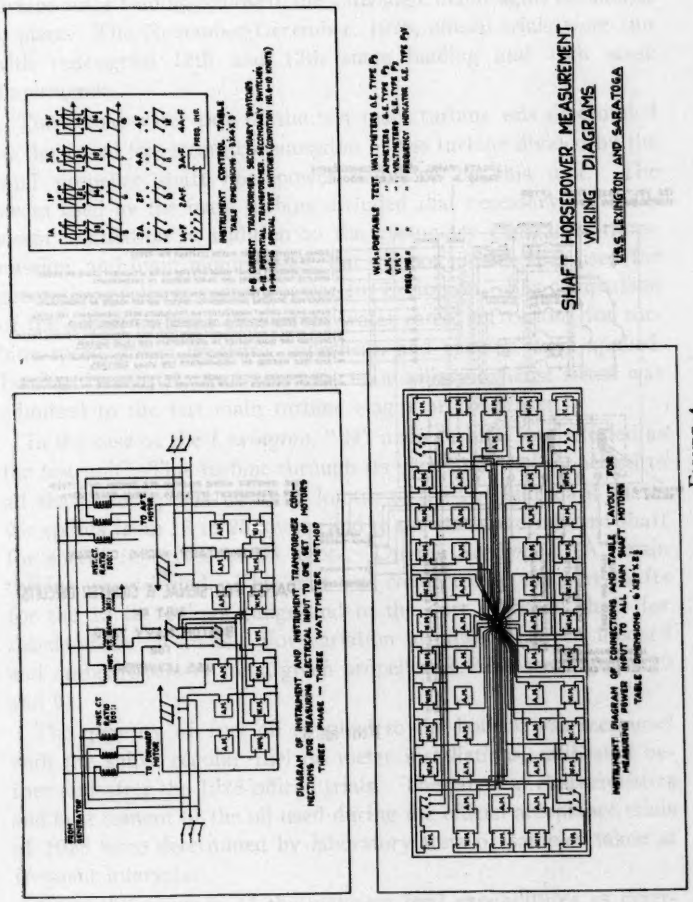

SHAFT HORSEPOWER MEASUREMENT
WIRING DIAGRAMS
U.S.S. LEXINGTON AND SARATOGA
INSTRUMENT CONTROL TABLE
DIAGRAM OF INSTRUMENT AND INSTRUMENT TRANSFORMER CONNECTIONS FOR MEASURING ELECTRICAL INPUT TO ONE SET OF MOTORS
THREE PHASE — THREE WATTMETER METHOD
INSTRUMENT DIAGRAM OF CONNECTIONS AND TABLE LAYOUT FOR MEASURING SHAFT HORSEPOWER, ALL MAIN PARTY MOTORS

FIGURE 4.

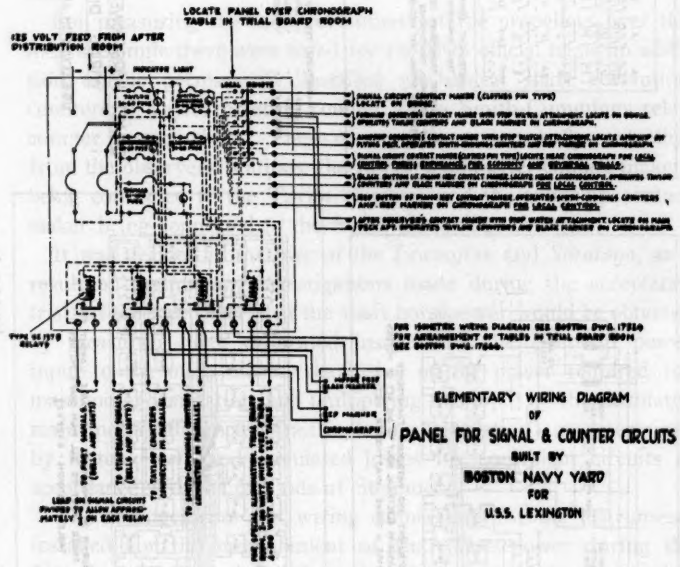

LOCATE PANEL OVER CHRONOGRAPH TABLE IN TRIAL BOARD ROOM
125 VOLT FEED FROM AFTER DISTRIBUTION ROOM
INDICATOR LIGHT
LOCAL SIGNALS
TYPE 35 STEP RELAY
FOR ISOMETRIC WIRING DIAGRAM SEE BOSTON DWG. 17250
FOR ARRANGEMENT OF TABLES IN TRIAL BOARD ROOM SEE BOSTON DWG. 17660.
TO FIRE MARSHALL (BELLS AND LIGHTS)
TO STEERSMAN SIGNAL (BELLS AND LIGHTS)
TO TAYLOR COUNTERS CONNECTED IN PARALLEL
TO SMITH-DIRECTOR COUNTER CONNECTED IN PARALLEL
LEAD ON SIGNAL CIRCUIT PASSING TO ALARM APPARATUS ON EACH RELAY
FOR SPECIAL ILLUMINATING TRIAL BOARD 3-WIRE 110 VOLT FEED FROM AFTER DISTRIBUTION ROOM
ELEMENTARY WIRING DIAGRAM
OF
PANEL FOR SIGNAL & COUNTER CIRCUITS
BUILT BY
BOSTON NAVY YARD
FOR
U.S.S. LEXINGTON

FIGURE 5.

sumption runs by means of calibrated condensate meters. The trials of May-June, 1928, were run with the 12th and 13th stage turbine rotor blading removed, the 13th stage diaphragms remaining in place. The November-December, 1928, official trials were run with redesigned 12th and 13th stage blading and 13th stage diaphragms.

The actual water rate of the test main turbine was determined by the actual live steam consumption of the turbine divided by the total propeller shaft horsepower delivered by this unit. The steam used by the main turbine included that necessary for main motor ventilation in addition to the customary generator, transmission, and main motor losses, but did not include that used for generation of direct current power for excitation. For comparison of the actual with the guaranteed water rates, corrections for turbine speed, steam pressure, superheat, and vacuum were applied. During the official water rate runs no auxiliary exhaust steam was admitted to the test main turbine stages or condenser.

In the case of the *Lexington,* "D" main turbine was selected as the test unit. This turbine through its generator was connected to all shafts for speeds up to 19 knots; to the two starboard shafts for speeds from 19 to 25 knots; and to the starboard inboard shaft for speeds in excess of 25 knots. On the *Saratoga,* "A" main turbine was selected. This unit was connected to the port shafts for the 19 to 25 knot range and to the port outboard shaft for speeds above 25 knots. For variation in power between inboard and outboard shafts for a given propeller R.P.M. see Figures 20 and 21.

The quantity of fuel oil supplied to the boilers was measured with the ships' regular fuel oil meter installations, calibrated before and after the 1928 official trials. The physical characteristics and heat content of the oil used during the official acceptance trials of 1928 were determined by laboratory tests of samples taken at frequent intervals.

Since the accuracy of the make-up feed expenditures as determined from tank soundings is largely dependent, in the case of these vessels, on the duration of the run, attention is invited to the notes included in the tabular data for runs of short duration.

The steam pressures and temperatures at the throttles and the exhaust pressures of the test main turbines were measured by means of calibrated instruments installed in addition to the regular equipment of the ships. All other pressures and temperatures were measured with the ships' standard equipments.

The gages connected to the auxiliary exhaust admission belts of the main turbines (8th and 11th stages) indicated the true stage pressures only when no auxiliary exhaust was being admitted. When auxiliary exhaust steam was admitted to a stage the gage indicated the pressure ahead of the auxiliary exhaust admission orifices.

The main feed water heaters were not fitted with feed inlet thermometers. Where feed heater inlet temperatures are shown in the tabular data they were calculated from the feed tank temperatures weighted for the estimated division of flow of the total steam condensate.

The quality of the auxiliary exhaust steam was not determined directly. However, the superheat of this steam at the evaporator 1st effect tube nest header was determined with the permanent shipboard instruments and from these values (Item 138 of Tabular Data) the quality of the auxiliary exhaust at the forward engine room can be calculated from steam tables.

Revolutions per minute of auxiliary machinery were measured with portable tachometers.

Uptake temperatures were not obtained on either vessel at any time. Analyses of flue gas samples from one boiler of the *Lexington* were made during the 1928 official trials.

The direct current power generation and distribution was measured with the ships' regular switchboard instruments.

For the official standardization trials all observations were taken at the beginning and at the end of each run over the measured mile, except the readings of the electrical instruments for determining the official shaft horsepower which were taken as frequently as the instruments could be accurately read and recorded. For the official water rate and fuel consumption runs observations were taken at thirty minute intervals, except for the shaft horsepower instruments which were read at frequent intervals. Frequency of observations for the data tabulated for unofficial runs

MAY 23

TABLE OF PROPELLING MACHINERY REQUIRED IN SERVICE ON CONTRACT TRIALS

TRIAL CONDITIONS	4000 S.H.P.(ABT 200 R.P.M.)			440 S.H.P.(ABT 15.5 K.)			220 S.H.P.(ABT 10.5 K.)			125 S.H.P.(ABT 7.5 K.)			275 S.H.P.(ABT 10.0 K.)			4000 S.H.P.(ABT 20.0 K.)			REMARKS
ITEM	NO. OF UNITS	S.H.P.	PRESS. (LBS. PER SQ. IN.)	NO. OF UNITS	S.H.P.	PRESS. (LBS. PER SQ. IN.)	NO. OF UNITS	S.H.P.	PRESS. (LBS. PER SQ. IN.)	NO. OF UNITS	S.H.P.	PRESS. (LBS. PER SQ. IN.)	NO. OF UNITS	S.H.P.	PRESS. (LBS. PER SQ. IN.)	NO. OF UNITS	S.H.P.	PRESS. (LBS. PER SQ. IN.)	
ENGINE ROOM EQUIPMENT																			
MAIN TURBO GENERATORS	1	850		1	850		1	1700		2	1250		4	2500		4	1750		
MAIN GEN. BOOSTER SETS	1			1			1			2			4			4			
MAIN CIRCULATING PUMPS	1	550		1	750		1	850		4	500		4	950		8	1000		
MAIN CONDENSATE PUMPS	1	1250		1	1250		1	1250		2	1250		4	1250		4	1250		
MAIN EJECTOR AIR PUMPS	2			2			2			4			8			8			
MAIN LUBRICATING OIL PUMPS	1	750	90"	1	750	90"	1	750	90"	2	750	90"	4	750	90"	4	750	90"	
750 K.W. GENERATOR SETS	2	800		2	800		2	800		2	800		3	800		3	800		
DYNAMO CIRCULATING PUMPS	2	975		2	975		2	975		2	1150		3	975		3	975		
DYNAMO CONDENSATE PUMPS	2	1250		2	1250		2	1250		2	1250		3	1250		3	1250		
DYNAMO EJECTOR AIR PUMPS	2			2			2			2			3			3			
MAIN FEED PUMPS	1	2700	350"	1	2700	350"	1	2700	350"	2	2700	340"	4	2250	350"	4	2700	350"	"3 HAND VALVE OPEN @ 18.0 K., 20.7 K., & F.P.
AUX. & RESERVE FEED PUMPS																			AS NECESSARY
MAIN FIRE & BILGE PUMPS							1												AS NECESSARY
AIR COMP. DIS. PUMP (WITH SERV. PUMP)	1	1250		1	1250		1	1250		2	1250		4	1250		4	1250		EMERGENCY SERVICE ONLY
HYDRAULIC VACUUM PUMPS	1			1			1			2			4			4			EMERGENCY SERVICE ONLY
FUEL OIL SERVICE PUMPS	1	240	175"	1	250	175"	1	250	270"	2	750	300"	2	1250	300"	4	1250	300"	"3 HAND VALVE OPEN AT ALL SPEEDS ABOVE 18.0 K.
FUEL OIL BOOSTER PUMPS	1	2½	50"	1	3½	50"	1	4½	50"	2	6½	60"	2	12½	50"	3	15	50"	
STEAM DRIVEN AIR COMPRESSORS																			AS NECESSARY
TURBINE GLAND EXHAUSTERS	1	2250		1	2250		1	2250		2	2250		4	2250		4	2250		
AIR COOLER SECTIONS FOR GEN.	16			16			16			16			16			16			
FEED HEATERS	1			1			1			2			3			4			
LUB. OIL COOLERS - MAIN TURBO GEN.	1			1			1			4			6			8			
LUB. OIL COOLERS - 750 K.W. SETS	2			2			2			2			3			3			
F.O. RECIRC. SYSTEM - F.O. RECIRC. PUMPS	1			1			1			1			2			3			IF REQUIRED
F.O. RECIRC. HEATERS	1			1			1			1			2			3			IF REQUIRED
BOILER ROOMS																			
BOILERS	2		295"	4		295"	4		295"	12		295"	16		295"	16		295"	
SUPERHTRS PER BOILER	1 Pass		260"	1 Pass		260"	1 Pass		260"	1 Pass		100"	1 Pass		295"	1 Pass		260"	
F.O. SERVICE HEATERS	2			4			4			12			16			16			
FORCED DRAFT BLOWERS	2	900	5.5"	4	900	2.2"	4	1000	4.0	12	900	3.3"	16	1000	4.0"	32	1250	6.0	PRESS. FIGURES IN INCHES OF WATER. ONE HAND VALVE OPEN @ 18.0 K., 20.7 K. & F.P.

BOILERS	2		250	4		250	4		250	12		250	16		250	16		250	
BURNERS PER BOILER	2 Sprays		100	2 Sprays		100	2 Sprays		225	2 Sprays		180	2 Sprays		250	2 Sprays		225	
FUEL OIL SERVICE HEATERS	2			4			4			12			16			16			
FORCED DRAFT BLOWERS	2	900	5.5	4	900	5.5	4	1000	4.0	12	900	5.5	16	1000	4.0	32	1000	5.0	SMALL PRESS. IN INCHES OF WATER. ONE MAIN VALVE OPEN @ 10.6K, 30.7K & 51K
MOTOR ROOMS																			
MAIN MOTORS	4	96		4	144		4	162		4	216		8	275		8	317		
DOUBLE FAN EXHAUSTERS	4			4			4			4			8			8			
DOUBLE FAN EXHAUSTERS										4			4			4			
LUBRICATING OIL PUMPS	4	700	2½	4	700	2½	4	700	2½	4	700	2½	4	900	2½	4	900	2½	
OIL & AIR COOLING CIRC. PUMPS	2	1000		2	1000		2	1000		2	1000		2	1000		2	1000		
AIR COOLER SECTIONS PER MOTOR	12			12			12			12			12			12			
DISTILLING PLANT	1 COMPLETE UNIT			1 COMPLETE UNIT			1 COMPLETE UNIT			1 COMPLETE UNIT			1 COMPLETE UNIT						ASS. ENGINES ABOUT 110, 0 INCH. VACUUM ABOUT 26"

GENERAL NOTES

VALUES FOR R.P.M. & DISCHARGE PRESSURE ARE ESTIMATED
DISCHARGE PRESSURE IN LBS. PER SQUARE INCH UNLESS OTHERWISE NOTED

ESTIMATED TABLE OF PROPELLING MACHINERY REQUIRED IN SERVICE ON CONTRACT TRIALS

U.S.S. LEXINGTON

TABLE 1

TABLE 5

SUMMARY OF ESTIMATED STEAM AND FUEL CONSUMPTIONS

PREPARED BY

GENERAL ELECTRIC COMPANY FOR U.S.S. KENTUCKY & SARATOGA

OPERATING CONDITIONS:								
Knots	10	16	16	16	26	26	30	33.7
Total shaft horsepower	4000	15500	28000	28000	97000	97000	181800	180000
Propeller r.p.m.	91	144	162	168	226	226	274	317
Steam conditions, at main turbines, press.G.-sqnht.°F-in.Hg.abs.	x----------			265 - 80 - 1½				----------x 265-80-2
Water rate, main turbines without aux.exhaust cdn.,lbs./s.hp.hr.	14.75	13.0	11.2	13.75	11.6	12.1	10.8	11.2
Number of main turbo-generators in operation	1	1	1	2	2	4	4	4
Number of main motors in operation	4	4	4	4	4	8	8	8
Main motor pole connection	44	44	44	22	22	22	22	22
STEAM CONSUMPTION:								
Main turbines without auxiliary exhaust steam admission,lbs./hr.	44975	248000	444400	2681200	4588000	8697000	19768000	28218000
Idle steam correction for aux. exh. steam admission, lbs./ hr.	7125	14750	26800	82700	47600	97200	84000	134000
Net idle steam to throttle, turbines operating with auxiliary exhaust admission, lbs. / hr.	52080	148150	219600	275400	606100	686800	1012800	1958080
Auxiliary direct current turbo-generators, lbs. / hr.	30000	30700	30900	31000	32000	36000	37750	38400
Steam auxiliaries other than d.c. turbo-generators, lbs. / hr.	42366	70388	108768	110804	207086	297878	548648	648648
Steam for miscellaneous purposes, lbs. / hr.	4840	5460	5750	5900	8080	9650	10650	14650
Total steam, all purposes(total evaporation), lbs. / hr.	127066	466648	851067	414884	867186	1826180	1965648	2486288
Total auxiliary steam(total steam less amt to m.turb.),lbs./hr.	87686	98648	151467	158484	260286	451088	861048	968688
Auxiliary steam, per cent of total steam	68.4	37.0	37.8	38.6	30.2	24.8	27.5	28.6
EVAPORATION PER POUND OF OIL:								
Total heat one pound of steam at boiler(200 lbs.abs.,60°sqnht.),Btu.	x---- - ----------			1246.7				----------x
Feed water temperature, °F.	x----------			220				----------x
Heat to be supplied to evaporate one pound of water, B.t.u.	x----------			1026.7				----------x
Number of boilers in operation	2	2	4	6	10	10	14	16
Average boiler efficiency(assumed to be 85 below expected test results), per cent	76	76	76	76	76	76	75.5	75
Heat value of fuel oil, B.t.u. per pound	x----------			19000				----------x
Actual evaporation per pound of oil	14.07	14.07	14.07	14.07	14.07	14.07	13.99	13.69

Average boiler efficiency(assumed to be 85 below expected test results), per cent	76	76	76	76	76	76	75.5	75
Heat value of fuel oil, B.t.u. per pound	19500							
Actual evaporation per pound of oil	16.07	16.07	16.07	16.07	16.07	16.07	15.97	15.69
Equivalent evaporation per pound of oil	15.87	15.87	15.87	15.87	15.87	15.87	15.16	14.66
*Fuel consumption(19500 B.t.u. Oil):								
Pounds of fuel oil per hour, all purposes	9069	19020	28788	39410	50300	62520	106000	138000
Tons of fuel oil per day	96.8	204	307	416	540	670	1070	1480
Pounds of fuel oil per shaft horsepower hour, all purposes	0.000	1.000	1.120	1.340	1.007	2.101	.904	1.000
Horsepower per ton of oil	2.47	1.97	1.66	1.36	.999	.895	.673	.600
Tons of oil per knot	.404	.900	.610	.721	1.00	1.12	1.49	2.49
Pounds of oil per knot	904	1107	1506	1654	2004	2012	2000	5400

- - - - -

*Estimates as originally prepared were based on 18000 B.t.u. oil; for the purposes of this article the estimated fuel consumptions have been corrected to 19500 B.t.u. oil.

TABLE 2

TABLE 6

ESTIMATE OF AUXILIARY STEAM AND ELECTRICAL LOADS

PREPARED BY

GENERAL ELECTRIC COMPANY FOR U.S.S. LEXINGTON & SARATOGA

Knots S.hp.	10 4800	14 16000	18 28000	18 28000	33 97000	33 97000	33 101000	33.7 105000
STEAM AUXILIARIES								
(DIRECT, WITH S.E. Back-Pressure Turbines)								
ON DC SWITCHBOARD:								
Main circulating pumps	9000	23400	46000	37440	105000	82000	171000	168000
Main condensate pumps	5000	8500	8000	4000	5000	10000	10600	11000
Main feed pumps	8500	11900	18100	21000	34800	34000	83600	79000
Dynamo cond.circ.pumps	5000	5000	5000	5000	5000	5000	5000	5000
Dyn.cond.condens.pumps	1000	1000	1000	1000	1000	1000	1000	1000
Forced lub.oil pumps	1200	1200	1200	2500	2500	2700	2700	2700
Oil & air cooler cir.p.	1000	2400	2400	2400	2400	4800	11200	10800
Fuel oil service pumps	1200	1600	2000	2000	4700	2000	7800	12000
F.O.recirc.pumps	500	500	500	500	500	500	500	500
F.O.booster pumps	200	200	200	200	200	200	200	200
Forced draft blowers	6700	12000	10600	24000	28000	40400	70000	251000
Air ejectors,main	516	1032	1662	2064	3096	4128	6192	6192
Air ejectors,dyn.cond.	730	730	730	730	730	730	730	730
Air ejectors, evap.	250	250	250	250	250	250	250	250
Fire pumps	-	-	-	-	-	-	-	-
Bilge pumps	1000	1000	1000	1000	1000	1000	1000	1000
Air compressors	1000	1000	1000	1000	1000	1000	1000	1000
Totals	42596	70886	108702	116554	297096	197978	548648	548942
A.C.TURBO-GENERATORS								
(DIRECT, WITH S.E. Back-Pressure Turbines)								
ON AC SWITCHBOARD No. 1	1200	2800	4000	4000	4800	4800	8500	8400
Generators No's.2 & 3	10800	12000	12000	19800	19480	21800	21660	22000
Totals	20400	24700	28000	23800	24280	26600	29780	30400
STEAM FOR MISCELLANEOUS								
(DIRECT, WITH S.E. Back-Pressure Turbines)								
Heating fuel oil	900	1230	1890	1662	3770	4600	6480	10700
Galley,laundry, hot water service	1780	1780	1780	1780	1780	1780	1780	1780

Knots S.hp.	10 4800	14 16000	18 28000	18 28000	33 97000	33 97000	33 101000	33.7 105000
AUXILIARY GENERATOR NO. 1---KW:								
Excitation	41	92	118	118	214	220	226	496
AUXILIARY GEN'S. NO.2 & 3---KW:								
Fresh water pumps	5	5	5	5	5	5	5	5
Incinerator	8	8	8	8	8	8	8	8
Elevators	10	10	10	10	10	10	10	10
Motor room lub.oil pumps	72	72	72	72	72	72	72	72
Mot.rm.circoil cooler circ.pump.	100	100	100	100	100	200	200	200
Refrigeration	48	48	48	48	48	48	48	48
Air compressors, S.E. & terp.	44	44	44	44	44	44	44	44
Steering gear	28	40	48	48	88	88	90	100
Lights,brwkst fans & elec.htrs.	200	200	200	200	200	200	200	200
Interior communication	6	6	6	6	6	6	6	6
Battery charging	5	5	5	5	5	5	5	5
Ship's ventilation	200	200	200	200	200	200	200	200
Galley,laundry,& bakery	150	150	150	150	150	150	150	150
Workshops	50	50	50	50	50	50	50	50
Secondary dredges	35	35	35	35	35	35	35	35
Sanitary pumps	30	30	30	30	30	30	30	30
Evaporator pumps	40	40	40	40	40	40	40	40
Miscellaneous	80	80	80	75	75	100	100	100
Total generator No.1	41	92	118	118	214	220	226	496
Total generators No.2 & 3	1068	1068	1068	1180	1180	1284	1288	1288
Total d.c. power	1180	1188	1307	1284	1347	1488	1488	1779

[redacted] FOR AUXILIARIES
[redacted] STEAM

Heating fuel oil	960	1210	1500	1680	2970	4660	6480	10700
Galley, laundry, hot water service	1780	1780	1780	1780	1780	1780	1780	1780
Heating vessel	2000	2000	2000	2000	2000	2000	2000	2000
Totals	4640	5490	5780	6460	6750	8440	10260	14480

[redacted] AUXILIARY
[redacted] STEAM---67920 90048 121467 139404 [redacted] 321040 341048 [redacted]

[redacted] OF AUXILIARY EXHAUST STEAM:

Total exh. steam other than d.e. gen.lbs./hr.	62360	70068	108782	110004	207006	197278	242648	242648
Less air ejectors	1016	2032	2032	2044	4066	5188	7188	7188
Net available at 10 lbs. G.,lbs./hr.	60080	68068	106750	107960	202000	192180	234460	234760
Used for evap.lbs/hr.	4370	4660	5180	5870	7120	7750	8060	9480
Balance, lbs./ hr.	56480	63480	96630	101970	195880	184400	227060	227150
To closed feed water heaters, lbs. / hr.	15000	32000	43000	50000	100000	100000	170000	320000
By-pass to main turb. lbs. / hr.	51480	51480	94130	51970	97880	81400	197060	227150
Stage admitted to	8	8	8	8	8	8	8	10
Stage press.,lbs/sq.in.abs. 8th stage	5.0	10.7	14.1	8.8	17.0	9.8	15.7	20.4
10th stage	1.8	4.3	5.7	3.8	10.0	4.0	6.8	11.4
H.hp. developed by exh. exhaust steam	488	1260	2360	1670	4120	3080	7780	9800
Correction in steam admitted at main throttle throttle in lbs thereof, lbs. / hr.	7188	16780	26260	22700	47480	37280	84200	118040

TABLE 8

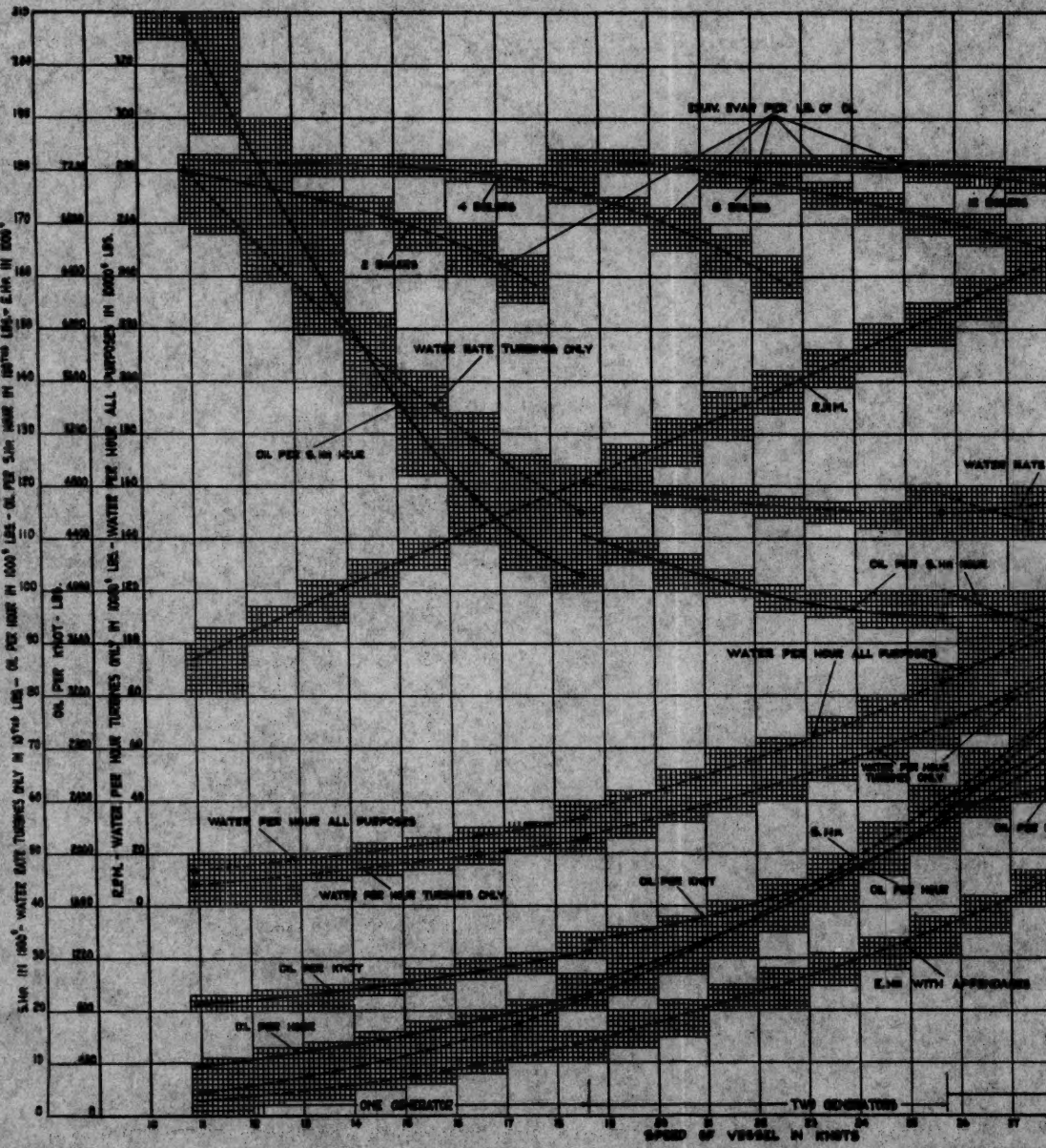
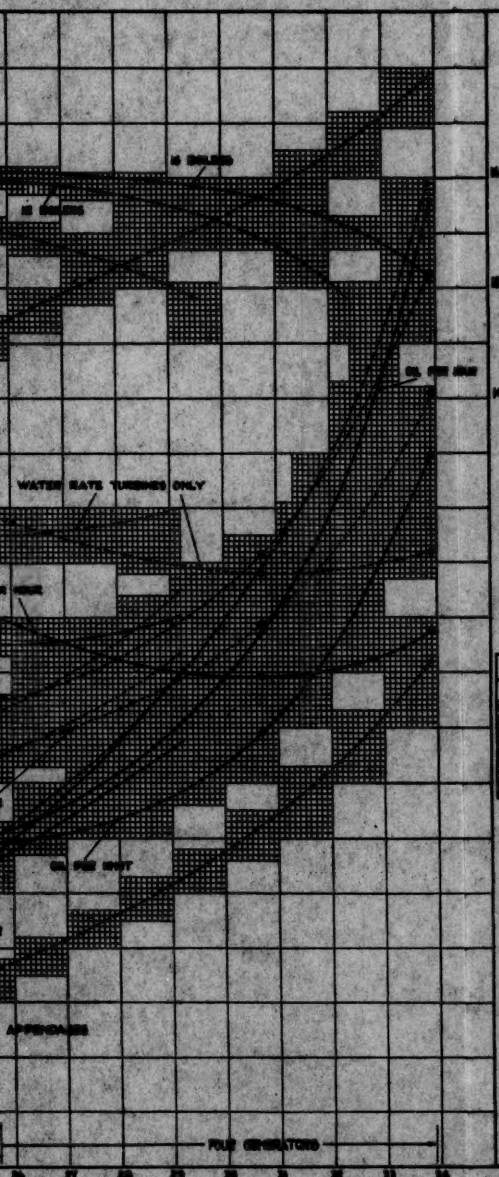

EQUIV. EVAP PER LB. OF OIL
4 BOILERS
3 BOILERS
2 BOILERS
WATER RATE TURBINES ONLY
S.H.P.
OIL PER S.H.P. HOUR
WATER RATE
OIL PER S.H.P. HOUR
WATER PER HOUR ALL PURPOSES
WATER PER HOUR TURBINES ONLY
WATER PER HOUR ALL PURPOSES
WATER PER HOUR TURBINES ONLY
S.H.P.
OIL PER KNOT
OIL PER HOUR
OIL PER KNOT
E.H.P. WITH APPENDAGES
OIL PER HOUR
ONE GENERATOR
TWO GENERATORS
SPEED OF VESSEL IN KNOTS
S.H.P. IN 1000's - WATER RATE TURBINES ONLY IN 10'ths LBS. - OIL PER HOUR IN 1000's LBS. - OIL PER S.H.P. HOUR IN 10'ths LBS. - E.H.P. IN 000's
OIL PER KNOT - LBS. - WATER PER HOUR TURBINES ONLY IN 000's LBS. - WATER PER HOUR ALL PURPOSES IN 000's LBS.
E.H.P. - WATER PER HOUR TURBINES ONLY IN 000's LBS. - WATER PER HOUR ALL PURPOSES IN 000's LBS.

FIGURE 6.



CONDITIONS

BOILER STEAM PRESSURE 265 # G 50° SUPERHEAT
STEAM CONDITIONS - AT TURBINE THROTTLE 265 # G 50° SUPERHEAT
VAC. @ 29.9 IN. - 28" AT ALL OTHER SPEEDS 28.9"
STEAM CONDITIONS FOR AUXILIARIES: AT THROTTLE VALVE - 250 # G 15° SUPERHEAT
BACKPRESS 10 # G
WATER RATES TURBINES ONLY USED IN CALCULATIONS ARE 6.5 LB. GUARANTEED.
B.T.U. PER LB. OIL 18200.
ESTIMATES ARE NET VALUES WITHOUT MARGINS. THEY INCLUDE OPERATION OF EVAPORATING PLANT AS REQUIRED BY CONTRACT, BUT DO NOT INCLUDE ALLOWANCE FOR FUTURE ELECTRIC SYSTEM FOR FIRE VENT. HEATERS AND RADIATORS.

CORRECTION FACTORS

	FOR 1 LB. INCREASE OR DECREASE IN TURBINE WATER RATE						FOR USE OF EVAPORATING SYSTEM						FOR USE OF VENT HEATERS & RADIATORS AT 0° F OUTSIDE TEMP.					

REFERENCES

A - VALUES FOR E.H.P. FROM EFFECTIVE HORSEPOWER CURVES MODEL # 2216. LENGTH 2232 ft. & C.O.R. # 10681. DISP. # 44-47 1000. DISPLACEMENT 33000 TONS DATED - JUNE 1ST 1925.

B - VALUES FOR S.H.P. & E.H.P. FROM CURVE OF ESTIMATED PROPELLER PERFORMANCE ON BUREAU DRAWING # B-CV-54-A. DATED NOV. 2, 1925

CONTRACT TRIAL
ESTIMATED PERFORMANCE
FOR
U.S. AIRPLANE CARRIER LEXINGTON.
PREPARED BY
BETHLEHEM SHIPBUILDING CORP. DEC. 23, 1927

FIGURE 6.

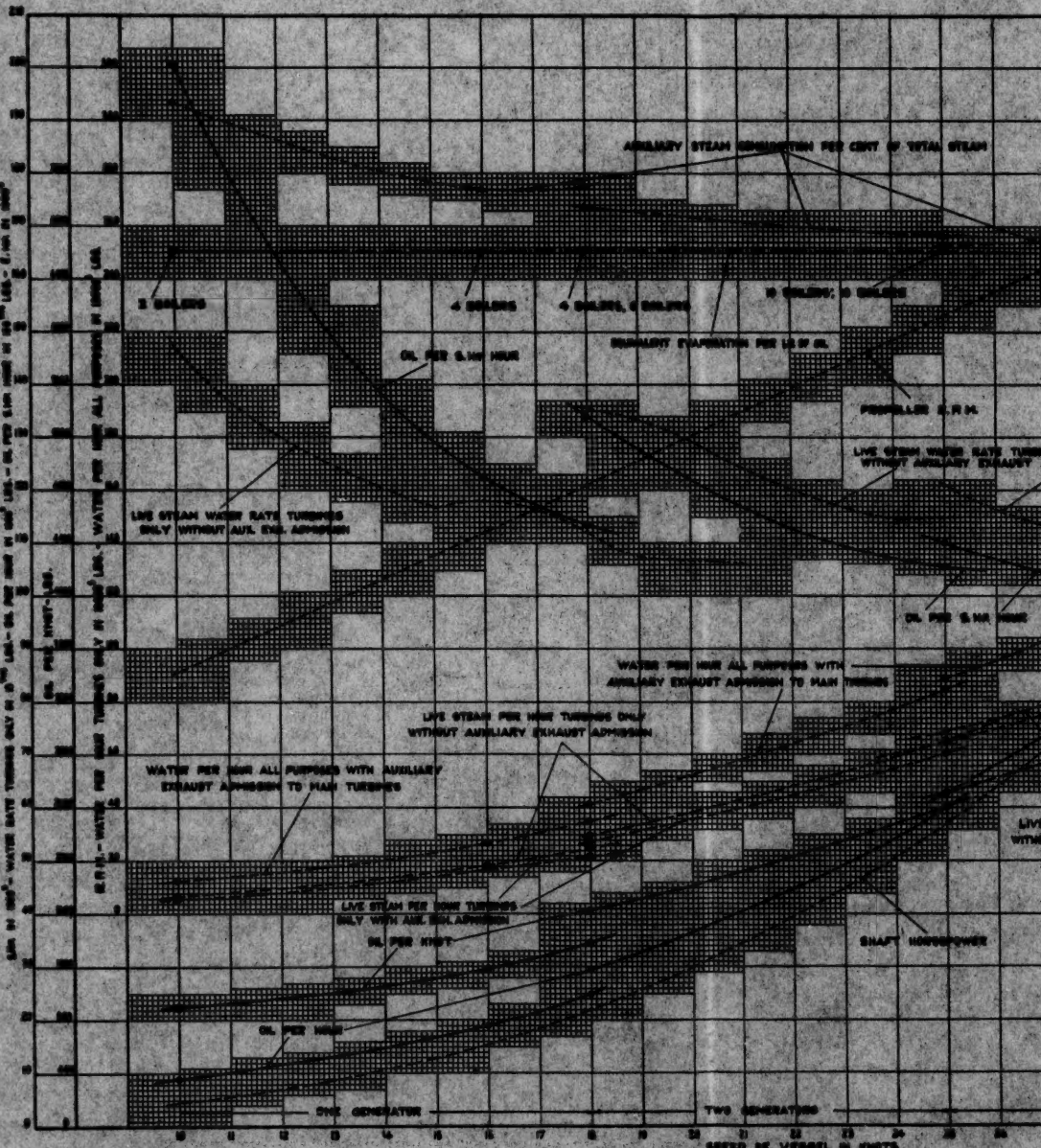

AUXILIARY STEAM CONSUMPTION PER CENT OF TOTAL STEAM
2 BOILERS
4 BOILERS
4 BOILERS, 2 BOILERS
EXHAUST EVAPORATION PER LB OF OIL
OIL PER S.H.P. HOUR
PROPELLER S.H.P.
LIVE STEAM WATER RATE TURBINES ONLY WITHOUT AUX. EXH. ADMISSION
WATER PER HOUR ALL PURPOSES WITH AUXILIARY EXHAUST ADMISSION TO MAIN TURBINES
LIVE STEAM PER HOUR TURBINES ONLY WITHOUT AUXILIARY EXHAUST ADMISSION
LIVE STEAM PER HOUR TURBINES ONLY WITH AUX. EXH. ADMISSION
OIL PER KNOT
OIL PER HOUR
SHAFT HORSEPOWER
ONE GENERATOR
TWO GENERATORS
SPEED OF VESSEL IN KNOTS

FIGURE 7

CONDITIONS AND REFERENCES

BOILER STEAM PRESSURE 235 lbs. G. 65° SUPERHEAT.

STEAM CONDITIONS AT TURBINE THROTTLE: 225 lbs. G. 25° SUPERHEAT.

VACUUM 28.5" AT ALL SPEEDS UP TO 70 AND INCREASING TO NORMAL GRADUALLY DECREASING TO 26.0" AT 90.7 KNOTS.

STEAM CONDITIONS FOR AUXILIARIES: AT THROTTLE VALVES 230 lbs. G. TEMP 30° F; BACK PRESSURE 20 lbs. G.

WATER RATES TURBINES ONLY USED IN CALCULATIONS ARE FROM GENERAL ELECTRIC COMPANY'S EXPECTED TEST RESULTS WITH THE TURBINES OPERATING WITHOUT AUXILIARY EXHAUST STEAM ADMISSION.

AUXILIARY EXHAUST (IN EXCESS OF THAT USED FOR FEED HEATING, DISTILLING PLANT, AND TURBINE GLANDS) ADMITTED TO LOW STAGES OF ALL MAIN TURBINES UP TO 20 KNOTS AND TO 25TH SPEED FOR HIGHER SPEEDS.

B.T.U. PER POUND OF OIL, 19000.

ESTIMATED ARE FOR NORMAL OPERATION UNDER SERVICE CONDITIONS WITH CLEAN BOTTOM AND GOOD WEATHER.

SPEED - S.H.P. & S.H.P. CURVES ARE FROM DATA FURNISHED THE GENERAL ELECTRIC COMPANY BY THE NAVY DEPARTMENT FOR THE MAIN PROPELLING MACHINERY.

ESTIMATED SERVICE PERFORMANCE
FOR
U. S. AIRPLANE CARRIERS LEXINGTON & SARATOGA
PREPARED FROM
GENERAL ELECTRIC COMPANY'S ESTIMATES OF PERFORMANCE MADE PRIOR TO COMMISSIONING.

FIGURE 7.

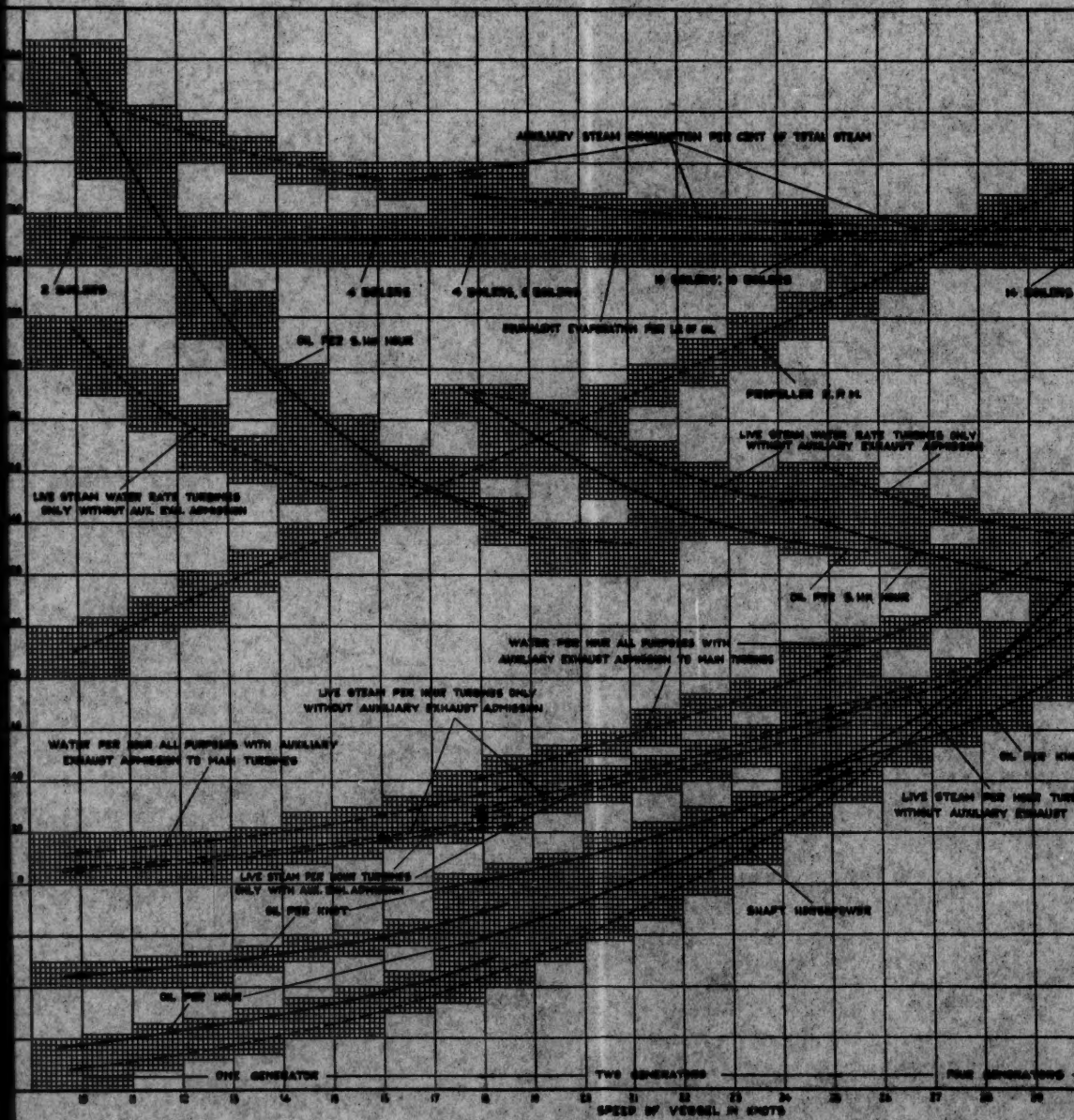
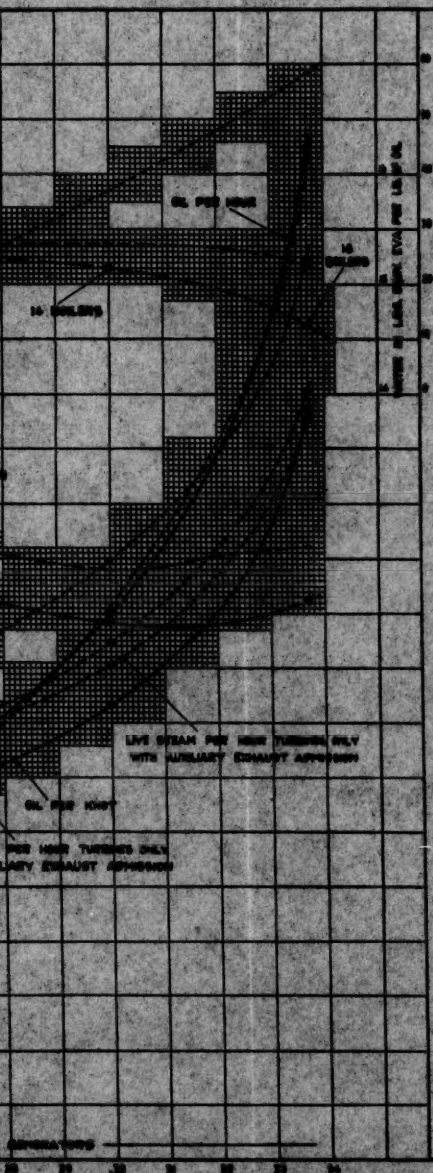

AUXILIARY STEAM CONSUMPTION PER CENT OF TOTAL STEAM
8 BOILERS
4 BOILERS
4 BOILERS, 6 BOILERS
10 BOILERS; 10 BOILERS
OIL PER S.H.P. HOUR
EXHAUST EVAPORATION PER LB OF OIL
PROPELLER R.P.M.
LIVE STEAM WATER RATE TURBINES ONLY WITHOUT AUXILIARY EXHAUST ADMISSION
LIVE STEAM WATER RATE TURBINES ONLY WITHOUT AUX. EXH. ADMISSION
OIL PER S.H.P. HOUR
WATER PER HOUR ALL PURPOSES WITH AUXILIARY EXHAUST ADMISSION TO MAIN TURBINES
LIVE STEAM PER HOUR TURBINES ONLY WITHOUT AUXILIARY EXHAUST ADMISSION
WATER PER HOUR ALL PURPOSES WITH AUXILIARY EXHAUST ADMISSION TO MAIN TURBINES
OIL PER KNOT
LIVE STEAM PER HOUR TURBINES ONLY WITH AUX. EXH. ADMISSION
OIL PER KNOT
SHAFT HORSEPOWER
OIL PER HOUR
ONE GENERATOR
TWO GENERATORS
SPEED OF VESSEL IN KNOTS

FIGURE 7.



CONDITIONS AND REFERENCES

BOILER STEAM PRESSURE 250# G. 60° SUPERHEAT.

STEAM CONDITIONS AT TURBINE THROTTLE: 240# G. 50° SUPERHEAT.

VACUUM 28.5" AT ALL SPEEDS UP TO AND INCLUDING 24 KNOTS GRADUALLY DECREASING TO 28.0" AT 28.7 KNOTS.

STEAM CONDITIONS FOR AUXILIARIES AT THROTTLE VALVES 230# G. SUPERHEAT 50° F; BACK PRESSURE 10# G.

WATER RATES TURBINES ONLY USED IN CALCULATIONS ARE FROM GENERAL ELECTRIC COMPANY'S EXPECTED TEST RESULTS WITH THE TURBINES OPERATING WITHOUT AUXILIARY EXHAUST STEAM ADMISSION.

AUXILIARY EXHAUST (IN EXCESS OF THAT USED FOR FEED HEATERS, DISTILLING PLANT, AND TURBINE GLANDS) ADMITTED TO 5TH STAGE OF ALL MAIN TURBINES UP TO 24 KNOTS AND TO 2ND STAGE FOR HIGHER SPEEDS.

B.T.U. PER POUND OF OIL, 19500.

ESTIMATES ARE FOR NORMAL OPERATION UNDER SERVICE CONDITIONS WITH CLEAN BOTTOM AND GOOD WEATHER.

SHAFT - H.P. vs. - R.P.M. CURVES ARE FROM DATA FURNISHED THE GENERAL ELECTRIC COMPANY BY THE NAVY DEPARTMENT FOR THE DESIGN OF THE MAIN PROPELLING MACHINERY.

ESTIMATED SERVICE PERFORMANCE

FOR

U. S. AIRPLANE CARRIERS LEXINGTON & SARATOGA

PREPARED FROM

GENERAL ELECTRIC COMPANY'S ESTIMATES OF PERFORMANCE MADE PRIOR TO COMMISSIONING.

varied from fifteen minute intervals for the preliminary shakedown runs to hourly intervals for the San Pedro-Honolulu run of the *Lexington.*

Unless otherwise noted the tabular data presented in this article is recorded as observed.

ESTIMATED PERFORMANCE.

Estimated steam and fuel consumptions for these vessels as prepared by the Bethlehem Shipbuilding Corporation for the *Lexington,* operating under *contract trial* conditions, and by the General Electric Company for the *Lexington* and *Saratoga,* operating under *service* conditions, made shortly before the vessels were commissioned, are shown in Tables 1-3 and Figures 6 and 7.

The estimate for the *Lexington* was prepared by the engineering staff of the shipyard as a guide for operation during the shakedown and official trial periods. Estimates are net values without margins and are based on 19,500 B.T.U. per pound fuel oil as called for in the fuel guarantees. Main turbine water rates are the values guaranteed by the General Electric Company, manufacturers of the main turbines. The knots-R.P.M.-S.H.P. curves used in this estimate are from the last approved estimated propeller performance curves released by the Navy Department, which, by comparison (Figures 6 and 7) can be seen to differ slightly from the earlier estimated propeller performance supplied the General Electric Company to which the main propelling machinery was designed and on which the General Electric estimated performance for the *Lexington* and *Saratoga* is based.

The General Electric Company's estimate for the *Lexington* and *Saratoga* was made to determine the quantity and quality of exhaust steam admitted to the main turbines under service conditions and to provide the operating personnel of the ships with a detailed analysis of the steam and auxiliary electrical load distribution together with estimates of fuel consumption as an aid in the attainment of high operating efficiencies. The accumulation of the necessary data to complete this analysis and the selection of the normal operating efficiencies was obtained with the assistance of the engineering personnel assigned to the fitting-out of the two vessels, the manufacturers of the main propelling machin-

ery, auxiliaries, and boiler room equipment, and the technical staffs of the shipyards.

The General Electric analysis is based on operating efficiencies of boilers and machinery expected to be maintained in normal service with clean bottom, good weather, negligible sea, and good steering. A heat content of 19,000 B.T.U. per pound of fuel oil was originally used in this analysis, but in order to permit more ready comparison with the Bethlehem Shipbuilding Corporation's estimate and the guarantee values the analysis has been revised to 19,500 B.T.U. oil. The steam consumption of the main turbines was derived from a curve of expected test results based on the calculated design values for the turbines operating without auxiliary exhaust steam admission, a small margin for unseen variations or contingencies being added, and from calculated values of the auxiliary exhaust steam admission to the main turbines, deduction in the steam flow through the main turbine throttle being made in accordance with the power produced by the auxiliary exhaust admitted to the turbines. The steam consumptions of the steam driven auxiliaries were obtained from test results made by the Naval Engineering Experimental Station at Annapolis or the various manufacturers of the equipment, due allowance being made for variation in speed and power to meet specific operating conditions as would be encountered in normal service. Steam for heating the vessel, for the galley, laundry, hot water service, and the heating of fuel oil, is lumped as "steam for other purposes," the steam for heating fuel oil being calculated, whereas the other items were derived from records of other Naval vessels corrected for the size of vessel, type, and number in crew. Priority in the distribution of the auxiliary exhaust steam was given the feed water heaters, evaporators, and turbine glands, the balance being admitted to the main turbines. In this estimate boiler efficiencies five per cent less than expected test results were used. The more pertinent results of this analysis are shown in Tables 2 and 3 and Figure 7.

TABULATED SUMMARIES OF PERFORMANCE DATA.

Tables 4 to 8 inclusive contain summaries of the pertinent data obtained during the various preliminary trials or runs conducted by the ships' forces and those obtained during the official trials under the supervision of the Board of Inspection and Survey.

TABLE 6

TABULATED PERFORMANCE DATA

MAXIMUM PERFORMANCE, ORIGINAL TURBINE BLADING — **U. S. S. MISSOURI**

ITEM No.	ITEM	16 KNOT RUN	16 KNOT RUN	20 KNOT RUN	25 KNOT RUN
	GENERAL DATA:				
1	Speed, knots, from design ship. curve	10.5	16.6	19.9	25.4
	; from subsequent Nov.1945 stds.	10.4	17.1	20.5	25.3
2	Duration, hours	4	17	4	4
3	Time commencing run	0000	1900	1700	1500
4	Date, 1945	March 31st	March 29-30rd	March 31st	March 10th
5	Locality	Gulf of Mexico	Caribbean Sea	West Coast of Cent. America	Florida East Coast
6	Days out of dock	26	24	28	24
7	Displacement	Service	Service	Service	Service
8	Trim by the stern, at number equivalent	3'-9"	3'-6"	3'-8"	3'-6"
9	Sea	Slight	Slight	Smooth	Smooth
10	Relative wind - velocity, knots	7	18	14	27
	- bearing, degrees	NNW	SSW	W	0-15
11	Barometer, inches Hg.	30.28	30.10	29.87	30.08
12	Outside air temperature, degrees F.	72	76	68	78
13	Temperature of sea, degrees F.	78	80	68	77
	MACHINERY DET-ITT:				
21	Boilers	8(12,14,16)	4(12,14,15,16)	8(5,6,6,7,10)	8(1 to 8 incl.)
22	Main turbo-generators	1(B)	1(B)	1(A)	2(A & C)
23	Main motors	4(Pod)	4(Pod)	4(Pod)	4(Pod)
24	Main motor pole combination	44	44	44	22
25	Main motor vent fans, number of 48000 c.f.m.elements	4	4	7	4
	- fan motors, 125 hp., number of (1 fan per mtr.)	4	4	4	4
	- fan motors, 250 hp., number of (2 fans per mtr)	0	0	3	0
26	Forced draft blowers, number of	8	4	5	16
	- total hand valves open	0	1	7	32
	- burner air register opening, fraction by scale 8.5/12	5/12	5/12	4/12	9/12
28	Fuel oil service pumps, number of	2	2	2	2
29	Fuel oil heaters, number of	2	4	5	6
30	Fuel oil burners, number of	12-16	16	24	48+
30a	Size of sprayer plates	5000	5000	5000	5000
31	Main feed pumps, number of	2	2	2	2
	- total hand valves open	-	-	-	-
32	Main feed heaters, number of	2	2	2	2

32	Fuel oil burners, number of	18-24	18	24	48
32a	Size of equiper plant	5000	5000	5000	5000
33	Main feed pumps, number of	2	2	2	3
	- total hand valves open	-	-	-	-
33	Main feed heaters, number of	2	2	2	2
34	Main circulating pumps, number of	1	2	2	4
	- total hand valves open	0	0	0	0
35	Main air pump units, number of	1	2	2	3
	- total number of 2-stage ejectors	1	2		6
36	Main condensate pumps, number of	1	1	1	2
39	Main turbine air & oil cooler & hydraulic governor pump supply pump, number of	1	1	1	2
41	750 kw. d.c. turbo-generators, number of	2	2	2	3
46	Dynamo condensing plants, number of	2	2	2	3
47	Motor room air & oil cooler pumps, number of	2	2	2	4
49	Distilling plant,No.quadruple effect units	2	2	1	3
50	Refrigerating plant,No.of after plant compressors	2	2	2	3
50a	Heating System	Steam	Steam	Steam	Steam
50b	Other auxiliary machinery	Standard	Standard	Standard	Standard
56	Disposition of auxiliary exhaust steam:-				
	Number of turbines admitted to	1	2	1	3
	-stage to which admitted	6th	6th	6th	10th
	Number of feed heaters admitted to	2	2	2	2
	Number of main exh.trunks admitted to	0	0	0	0
	No.of aux.condensers admitted to	0	0	0	0
	No.distilling plant units admitted to	2	2	2	2
57	Disposition of drains:-				
	High pressure to	Feed Htrs.	Feed Htrs.	Feed Htrs.	Feed Htrs.
	Low pressure to	Feed tanks	Feed tanks	Feed tanks	Feed tanks
	Feed heater to	Feed tanks	Feed tanks	Feed tanks	Feed tanks
	1st effect evaporator to	Hyd. Cond.	Hyd. Cond.	Hyd. Cond.	Hyd. Cond.
	Fuel oil heater to	Feed tanks	Feed tanks	Feed tanks	Feed tanks

TABLE 3 (CONTINUED)

TABULATED PERFORMANCE DATA

STANDARD PERFORMANCE, ORIGINAL TURBINE BLADING

Item No.	Item	10 knot run	15 knot run	U. S. S. Neosho 10 knot run	U. S. S. Neosho 20 knot run
	FUEL OIL CHARACTERISTICS:				
81	Specific gravity, degrees A.P.I.	30	30	30617	30
82	Pounds per gallon at 60 deg. F.	7.770	7.770	7.86	7.770
83	Viscosity:---------- d deg. Engler at	150 °F	150 °F	170°F	150 °F
84	Heat content, B.t.u. per lb.as fired,estimated	18900	18900	18900	18900
	PRESSURES:				
91	Steam at boilers, lbs. per sq. in. gage	270	270	272	261
92	Steam in engine room- fwd. flat, lbs.per sq.in.g.	270	270	270	260
	- aft. flat, lbs. per sq.in.g.	---	260	---	251
94	Steam chest, main turbines, lbs.per sq.in.,gage	---	---	---	247(A) 238(C)
96	1st stage, main turbines, lbs.per sq. in.,gage	10.5	98	90	247(A) 1.27(C)
98	8th stage admission bell, main turbines, lbs. per sq. in. gage or inches Hg.vacuum	12"	8.5"	8.5"	8.5"(A) 8.5"(C)
97	10th stage admission bell, main turbines, lbs. per sq. in. gage or inches Hg.vacuum	25"	25"	14"	10.5"(A) ---
96	Main turbine exhaust, inches Hg. vacuum	28.5	28.5	27.7	26.7(A) 26.1(C)
101	Auxiliary exhaust steam at aux. turb.,lbs.gage	15	14	15.5	15.6
102	Forced draft blower turbine steam bell, lbs.gage	150	145	154	150
104	Firerooom air, inches water	1.2	2.3	2.9	2.60
105	Fuel oil service pump discharge,lbs.per sq.in.g.	255	255	260	260
106	Fuel oil at burner manifolds,lbs.per sq.in.g.	250	245	250	260
107	Main feed pump discharge, lbs.per sq. in. g.	270	270	270	260
108	Main feed at boilers, lbs. per sq. in. g.	260	255	260	240
	TEMPERATURES:				
151	Steam from boiler superheaters, deg. F.	450	450	451	446
152	Superheat at boilers, deg. F.	10	10	10	30
156	Steam chest, main turbines, degrees F.	450	450	452	450(A) 450(C)
156	Superheat at aux.turb. steam chest, deg. F.	---	---	---	50(A) 50(C)
157	Main turbine exhaust, degrees F.	98	104	106	120(A) 115(C)
159	Fuel oil to pumps, degrees F.	87	91	90	90

136	Steam chest, main turbines, degrees F.	430	430	448	450(A) 450(S)
136	Superheat at exh.turb. steam chest, deg. F.	—	—	—	84(A) 88(S)
137	Main turbine exhaust, degrees F.	98	104	106	108(A) 116(S)
139	Fuel oil to pumps, degrees F.	87	91	98	90
140	Fuel oil at burner manifolds, degrees F.	141	138	137	134
141	°Feed water to feed heaters, degrees F.	130	130	116	—
142	Feed water from feed heaters, degrees F.	218	228	228	220
143	Main injection , degrees F.	79	80	88	77
144	Main condenser overboard discharge, degrees F.	88	93	90	91(A),88(S),91
145	Main condensate, degrees F.	84	92	88	86(A),88(S),94
149	Spares condenser overboard discharge, deg. F.	81	87	98	88
150	Spares condenser condensate, degrees F.	91	98	130	102
174	Forward main turbo-generator bearing, degrees F.	134	138	140	140
175	Aftmost main motor bearing, degrees F.	108	120	126	126
176	Forward propeller thrust bearing, degrees F.	108	97	98	100
178	Main generator air supply (from coolers),deg.F.	87	138	104	96
179	Main generator air discharge to coolers, deg.F.	134	150	140	122
181	Main motor air supply(room temperature), deg.F.	102	88	94	88
182	Main motor air discharge to coolers, deg. F.	112	118	116	140
186	Main generator stator coils, maximum, deg. F.	127	152	174	168
186**	Main generator field coils, maximum, deg. F.	—	—	200	200
188	Main motor stator coils, maximum, deg. F.	—	138	130	132

°Estimated from feed tank temperatures.

**Main control board generator field temperature indicator.

TABLE 4 (CONTINUED)

TABULATED PERFORMANCE DATA

ITEM NO.	ITEM	10 KNOT RUN	14 KNOT RUN	U. S. S. LEXINGTON 16 KNOT RUN	25 KNOT RUN
	REV. PER MIN. & DOUBLE STROKES:				
201	Main turbo-generators (by electric tachometer)	1025	1575	1750	1238
209	Fuel oil booster pumps(26)	--	--	--	10
210	Fuel oil service pumps	--	--	--	940
211	Forced draft blowers	--	--	--	1850
212	Main feed pumps	--	--	--	2980
216	Main circulating pumps	1000	--	--	1120
217	Main condensate pumps	--	--	--	1445
218	Dynamo condenser circulating pumps	--	--	--	1645
219	Dynamo condenser condensate pumps	--	--	--	2825
	MAIN PROPULSION ELECTRICAL:				
231	Main generator line volts	1525	2775	4775	3625
235	Main generator line amperes	1975	3235	3600	4700
236	Main generator line power factor	.71	.65	.65	.875
237	Main generator line indicated kilowatts, total	3000	13060	18975	51150
240	Main gen. line kw. from $\sqrt{3}$ x E x I x cos θ,total	3850	13750	19750	50900
248	Main motor line amperes:-				
	- motor 17	425	775	900	2325
	- motor 27	425	755	900	2350
	- motor 37	425	756	850	2295
	- motor 47	525	797	975	2400
249	Main generator field volts	76	118	150	161
251	Main generator field amperes	310	485	620	651
253	Excitation indicator	4.9	5.5	6.4	1.8
257	Main motor efficiency(from calculated eff.tables)	.90	.929	.928	.957
	SPEED, POWER, & PROPELLER R.P.M.:				
261	Speed of vessel, knots,corresponding to propeller r.p.m., from subsequent standardiz.of Nov.,1928	10.4	17.1	18.5	26.3
268	Total shaft horsepower - from control board instr.	5000	17800	24130	64860
	- from Nov.1928,stdz.curve	5000	18600	25700	67750
	- from Nov.1928 mechy.trials r.p.m.-s.hp.curve	4700	18400	23750	65600
267	R.p.m. of propellers, all shafts average	91.3	144.8	157	223.0

862	Speed of vessel, knots, corresponding to propeller r.p.m., from subsequent standardiz. of Nov., 1928	10.4		17.1	18.5	26.3
865	Total shaft horsepower - from control board instr.	5000		17800	24130	44560
	- from Nov.1928, stdz.curve	5000		18600	23700	47750
	- from Nov.1928 mechy.trials r.p.m.-s.hp.curve	4700		18400	23750	45000
867	R.p.m. of propellers, all shafts average	91.3		144.8	127	223.0
	AUXILIARY POWER:					
881	Average total d.c. power, 750 kw.set output, kw.	775		955	1000	1250
888	D.c. power for excitation incl.booster set losses, kw.	39		61	110	233
884	D.c. power for engine auxiliaries, kw.	75		139	90	198
888	D.c. power for ship's service power, kw.	588		655	675	675
896	D.c. power for ship's lighting, net, (lighting motor-generator output)	75		100	125	150
	FEED WATER:					
901	Make up feed, gallons per hour		Not known			
908	Distilled by evaporators, gallons per hour	2975		2985	2000	5400
	FUEL CONSUMPTION:					
911	Gallons at 60°F of fuel oil per hour, as fired	1170		2685	2089	6806
918	Pounds of fuel oil per hour, as fired	9100		19486	24509	68565
918a	Tons of fuel oil per hour, as fired	4.06		8.66	10.61	30.61
915	Lbs. of fuel per hr. per cu.ft. furnace vol., as fired	2.66		4.58	4.97	8.09
914	Lbs. of fuel oil per hr. per sq. ft. of water heating surface, as fired	.870		.451	.450	.761
918	Lbs. of fuel oil per shaft hp.hour, as fired	1.88		1.09	1.004	1.068
	EFFICIENCY:					
981**	Overall efficiency, fuel to shaft, all purposes	.076		.184	.126	.131

TABLE 4
SHEET 3

*Observed data from ship's switchboard instruments

**Based on fuel oil heat content (estimated) of 18000 B.t.u. per pound.

FANTASTIC FUTURE AGRICULTURAL MACHINERY DEALERSHIP

TABLE 1

FARM EQUIPMENT SALES DATA

	No. main circuit breaker - WH / FC	3/0	3/0	3/0	3/0	3/0	3/0	1	1	0	1
	No. blowers condensers - WH / FC	1/1	1½/1	1/1	0/1	1/1	1/1				
	No. distilling plant units - WH / FC										
87	Disposition of drains:-										
	High pressure to -	Ft.Rtr.	F.H.	F.H.	F.H.	F.H.	F.H.	F.H.	F.H.	F.H.	F.H.
	Low pressure to -	Ft.Tub.	F.H.	F.H.	F.H.	F.H.	F.H.	F.H.	F.H.	F.H.	F.H.
	Feed heater to -	Ft.Tub.	F.H.	F.H.	F.H.	F.H.	F.H.	F.H.	F.H.	--	F.H.
	1st effect evaporator to -	Eym.Cond.	D.C.	D.C.	D.C.	D.C.	D.C.	D.C.	D.C.		D.C.
	FUEL OIL CHARACTERISTICS:-										
81	Specific gravity, degrees A.P.I.	16.45	17.7	16.6	17.9	16.8	18.1	16.1	16.8	16.88	16.6
82	Pounds per gallon at 60 deg. F.	7.963	7.898	7.866	7.887	7.878	7.877	7.864	7.878	7.878	7.889
83	Viscosity: 4 deg. Engler @t	198°/	172°F	192°F	171°F	190°F	166°F	200°F	202°F	199°F	202°F
84	Water and sediment, per cent	0.6	1.7	0.6	1.8	0.6	2.6	0.6	0.6	0.6	0.6
85	Flash point, degrees F.	220	230	220	230	220	230	220	220	220	220
86	Source	Calif.		Calif.		Calif.		Calif.		Calif.	
87	Heat content, B.t.u. per pound, water free	18908	18936	18916	18956	18897	18974	18894	18900	18908	18900
88	Heat content, B.t.u. per pound, as fired	18736	18619	18618	18589	18776	18670	18776	18689	18768	18699
	TEMPERATURES:										
91	Steam at boiler, lb.per sq.in.&g. - WH / FC	278/278	278/276	271/271	278/278	271/274	266/267	275	284	276	288
92	*Steam in engine room,- Superheat ,lbs.steam,WH/FC	275/283	272/270	275/275	273/274	275/275	288 /282	276	271	270	268
	- after ,lbs.steam,WH/FC	276/283	280/278	270/270	274/280	270/275	282 /283	268	280	270	278
93	Steam at test main turbine throttle, S.H.Co. gage as read, lbs. per sq. in. Gage	260/260	268/263	260/255	268/266	259/256	260/263	230	239	241	
94	Steam chest, main turbines,lbs.per sq.in. Gage WH/FC	266/266	272/270	264/260	260/270	259/259	271/271	237	248	240	258
95	1st stage main turbines, lbs. steam, WH/FC	18.8/18.8	18/18	88/87	86/86	140/130	138/130	197	189	199	196
	-2nd turbine(no cnn.cnh.cdn.for Saratoga)								191	198	197
	-3rd turbine										197
	-4th turbine							194	--	206	198
	-5th turbine							224	196	199	196
	-6th turbine										
96	20th stage admission bell,lbs.steam or in.Hg.vac.:										
	-water rate run/fuel consumption run(1 burn.)	18"/2.9/	16/0.8/	0/0/	0.8"/2.0/	0.6/10/	0.6/2.0/	--	11.8/	12.5/	10.6/
	-1st turbine(no cnn.cnh.cdn.for Saratoga)	--	--	--	--	--	--	--	--	10.6/	12.5/
	-2nd turbine	--	--	--	--	--	--	10.8/	--	10.6/	0/0
	-3rd turbine	--	--	--	--	--	--	17.2/	11.2/	12.7/	11.7/
	-4th turbine(no cnn.cnh.cdn.for Lexington)	--	--	--	--	--	--				

*Lexington gages connect to dynamo flat cross connection in each engineroom; Saratoga gages to forward main steam cross connection of forward engineroom and after main steam cross connection of after engineroom.

Values listed are for total runs except where otherwise noted and where separated by /. ; values to left of / are for water rate run and to right of / for fuel consumption runs.

TABLE 8 (CONTINUED)

MAY - JUNE 1928 — TABULATED TRIAL DATA

SPECIAL ACCEPTANCE MACHINERY TRIALS — U. S. S. LEXINGTON & SARATOGA

ITEM NO.	ITEM	4500 SHP RUN LEX	4500 SHP RUN SARA	144 RPM RUN LEX	144 RPM RUN SARA	188 RPM RUN LEX	188 RPM RUN SARA	[illegible] RPM RUN LEX	[illegible] RPM RUN SARA	140000 SHP RUN LEX	140000 SHP RUN SARA
	PRESSURES (Continued):										
97	10th stage adm. belt, lbs. gage or in. Hg. vac. -single turbine runs, water rate/fuel consump.	27.5/26"	27"/26.7"	21"/20"	20"/19"	15"/14"	16"/15"	--	--	--	--
	-"A" turbine (no aux. exh. adm. for Saratoga)	--	--	--	--	--	--	--	10.0"	9"-4"	9.9"
	-"B" turbine - - -	--	--	--	--	--	--	--	--	4.5"-2.6"	3.0"
	-"C" turbine - - -	--	--	--	--	--	--	0.9"	--	6"	7.2"
	-"D" turbine (no aux. exh. adm. for Lexington)	--	--	--	--	--	--		5.0"	6.5"	3.4"
98	Main turbine exhaust, in. Hg. vac. (gage board) - WR/FC	29.2 /29	28/28.0	28.2/28	28.3/28.1	27.5/27.3	28.2/28.2	26.4	28.1	26.2	27.3"
99	Test main turbine exhaust, G. E. Co. absolute pressure gage, inches Hg. absolute - WR/FC	1.2/1.25	1.34/1.36	2.05/2.14	2.47/2.58	2.55/2.76	2.63/2.90	3.50	3.22	3.85	3.14
100	Dynamo condenser vacuum, inches Hg. (gage board)	29.2	29	29.1	29	29.7	28.9	29.4	29	27.8	29.2
101	Auxiliary exhaust steam at main turb., lbs. Gg. - WR/FC	12.5/12		12/12.5		10.7/10	11	11		11.7	14
104	Fireroom air, inches water	1.3	0.4	3.6	1.65	3.4	3.4	4.5	4.7	7.5	6.6
105	Fuel oil pump discharge, lbs. gage	280	250	265	300	310	200	315	200	295	310
106	Fuel oil at burner manifolds, lbs. gage	225	222	290	282	275	232	270	215	256	263
107	Main feed pump discharge, lbs. gage	375		375		360		360		380	
109	Main feed at boilers, lbs. gage	305	365	360	368	360	365	353	369	345	366
125	Main feed heater shell, lbs. gage	13		13		15		13.5		14	
	TEMPERATURES:										
131	Steam from boiler superheater, deg. F. - WR /FC	429/428	452/449	432/432	473/472	448/446	485/484	449	491	467	504
132	Superheat at boiler, deg. F. - WR / FC	14/14	36/35	19/19	56/57	35/32	67/66	35	71	56	81
133	Steam at test turbine throttle, deg. F. - WR /FC	421/424		431/430		449/448		441		457	
134	Superheat at test turb. throttle, deg. F. - WR /FC	12/15		25/22		42/40		39		54	
135	Main turbine steam chest, deg. F. - WR /FC	425/425	442/442	435/435	460/464	447/447	478/476	430	485	455	505
136	Superheat at main turb. steam chest, deg. F. - WR/FC	13/13		25/23		30/30		26		52	
137	Main turbine exhaust, deg. F. - WR/FC	85/98	88/96	100/101	108/109	108/110	104/112	120	116	118	117
138	Evaporator 1st effect tube nest header (aux. exh.) °F	218		219		230		220		--	
	- superheat, °F.	Wet		Wet		10.6		1		--	
139	Fuel oil booster pump suction, deg. F.	85	80	74	88	74	82	77	94	65	97
140	Fuel oil to burners, deg. F.	195	180	184	185	189	180	190	180	188	179
141	Main feed heater inlet, deg. F.	145/110	121	106/103	128	140	137	108	146	109	135
142	Main feed heater outlet, deg. F.	230		236		226		232		219	

159 Fuel oil booster pump suction, deg. F.	86	80	74	80	74	82	77	94	68	87
160 Fuel oil to burners, deg. F.	198	186	184	185	189	180	190	180	186	179
161 Main feed heater inlet, deg. F.	145/110	121	104/103	128	140	127	108	144	109	126
162 Main feed heater outlet, deg. F.	226		228		226		232		219	
163 Main injection, deg. F.	64	81	64	89	64	68	64	68	68	64
164 Main condenser overboard discharge, deg. F.	70	60	68	78	83	81	88	86	71	84
165 Main condensate, deg. F.	78	78	88	99	98	104	98	107	86	106
169 Dynamo condenser overboard discharge, deg. F.	77	68	71	64	78	78	79	68	76	74
170 Dynamo condenser condensate, deg. F.	90	84	91	85	98	96	100	89	104	96
166 Lubricating oil to main tur-gen.oil coolers, °F.	114	108	106	122	120	120	118	120	110	
168 Lubricating oil to gen.motor & thrust oil coolers, °F	92	96	96	106	96	96	90	99	100	
171 Lubricating oil to 750 kw.set oil coolers, deg.F.	127	120	128	114	124	--	128	118	128	
176 Warmest main turbo-generator bearing, deg.F.	130	121	130	148	134	149	139	148	143	149
178 Warmest main motor bearing, deg. F.	104	95	106	109	106	107	102	110	117	112
178 Warmest propeller thrust bearing, deg. F.	101		102		103		100		103	
177 Warmest 750 kw.set bearing, deg. F.	134	134	134	136	134	136	136	136	136	138
179 Main generator air discharge to coolers, °F	126	--	129	--	118	--	128	--	129	--
180 Main motor air discharge to coolers, deg. F.	99	89	99	99	114	97	127	101	111	101
186 Main generator stator coils, average, deg. F.	130	120	147	123	130	149	148	143	160	142
189 Main motor stator coils, average, deg. F.	130	120	132	128	134	133	131	140	139	127
190 Fireroom, 6 ft. above floor plates, deg. F.	102	84	--	82	91	86	92	84	88	86
191 Engineroom, 6ft. above floor plates, deg.F.	93		101		93		96		93	
192 Motor room, 6 ft. above floor plates, deg.F.	92	91	98	80	86	79	78	89	79	72

TABLE 5
[redacted]

MAY-JUNE 1928

SPECIAL ACCEPTANCE MACHINERY TRIALS

TABLE B (CONTINUED)

TABULATED TRIAL DATA

U. S. S. LEXINGTON & SARATOGA

Item No.	Item	4500 SHP Run LEX	4500 SHP Run SARA	144 RPM Run LEX	144 RPM Run SARA	168 RPM Run LEX	168 RPM Run SARA	224 RPM Run LEX	224 RPM Run SARA	140000 SHP Run LEX	140000 SHP Run SARA
	MAIN PROPULSION ELECTRICAL:										
[illegible]	Main generator line volts, average	1860	1860	3636	3600	4777	4566	3777	3693	4120	4732
[illegible]	Main generator line amperes, average	1810	2000	2170	2210	3694	3815	4891	4803	4130	3888
[illegible]	Main generator line power factor, average	.45	.61	.65	.663	.65	.67	.87	.89	.88	.861
[illegible]	Main generator line indicated kw., total	2500	3750	12630	15000	19655	21040	58300	58870	106097	111951
[illegible]	Main gen. line kw. from VJ E I cos θ, total	3654	3600	13687	14444	19877	20740	52860	54686	103748	110876
[illegible]	Main motor kw. input from a.kp. instruments, total	3700	3678	13860	14173	20455	20424	52097	52750	106753	109600
[illegible]	Main generator field volts, average	72	76	119	115	138	135	181	175	140	161
[illegible]	Main generator field amperes, average	506	373	496	586	583	600	727	700	575	666
[illegible]	Excitation indicator, average	9.1	9.4	5.3	7.8	4.4	5.8	3.8	3.6	4.0	6.4
[illegible]	Booster motor-generator "motor" and amps., avg.	182	160	300	180	200	188	110	185	190	109
[illegible]	Main motor efficiency, average (from tables)	.9006	.9002	.9277	.928	.9166	.9265	.9576	.9582	.9600	.9603
	SPEED, POWER, & PROPELLER R.P.M.										
261	Avg. speed of vessel in knots corres. to avg. prop. rpm	10.62	10.41	16.60	16.67	18.85	18.68	25.94	26.29	32.38	33.27
	- 2 hr. water rate run	10.31	10.33	16.56	16.64	18.88	18.70	--	--	--	--
	- 2 hr. fuel consumption run	10.78	--	16.62	--	18.60	--	--	--	--	--
	- 4 hr. fuel consumption run	--	10.48	--	16.69	--	18.67	--	--	--	--
262	Total shaft horsepower, all shafts average	4467	4447	17261	17574	25131	25181	67904	67612	136206	141149
	- 2 hr. water rate run	4266	4094	17230	17523	25198	25168	--	--	--	--
	- 2 hr. fuel consumption run	4467	--	17292	--	25104	--	--	--	--	--
	- 4 hr. fuel consumption run	--	4374	--	17600	--	25188	--	--	--	--
267	R.p.m. of propellers, all shafts average	92.01	86.31	144.15	144.17	162.04	162.09	225.36	226.04	285.90	286.43
	- 2 hr. water rate run	90.95	87.07	144.00	143.90	161.98	162.25	--	--	--	--
	- 2 hr. fuel consumption run	93.06	--	144.30	--	162.10	--	--	--	--	--
	- 4 hr. fuel consumption run	--	88.82	--	144.30	--	162.01	--	--	--	--
272	Propeller shaft hp. developed by test main turbine	4467	4447	17261	17574	25131	25181	34011	33772	38407	34830
	- 2 hr. water rate run	4266	4094	17230	17523	25198	25160	--	--	--	--
	- 2 hr. fuel consumption run	4467	--	17292	--	25104	--	--	--	--	--
	- 4 hr. fuel consumption run	--	4374	--	17600	--	25188	--	--	--	--
	AUXILIARY POWER:										
281	Average total d.c. power, kw.	825	992/941	1000	960/920	950	1020/963	1150	1212	1100	1430
282	Avg. d.c. power for excitation incl. booster losses, kw.	23	48/46	69	80/81	94	96/98	288	268	381	487
283	Avg. d.c. power absorbed by mn. gen. fields, net excit. kw.	22	29/28	62	60/61	80	81/82	208	242	329	428
284	Avg. d.c. power for engine auxiliaries, kw.	21	136/144	84	149/147	85	153/150	96	149	72	173
285	Avg. d.c. power for ship's service power, kw.	628	629/584	735	577/519	645	614/556	616	634	547	608

882	Avg. d.c. power for excitation incl. booster losses, kw.	45	48/44	69	80/81	94	96/98	388	382	381	487
886	Avg. d.c. power absorbed by an.gen.fields, net smiths.	28	29/28	62	60/61	89	81/82	255	242	329	426
884	Avg. d.c. power for engine auxiliaries, kw.	21	136/144	66	149/147	89	155/156	98	149	72	175
888	Avg. d.c. power for ship's service power, kw.	626	629/684	735	577/519	645	614/556	616	634	547	606
889	Avg. d.c. power for ship's lighting, net. (lighting motor-generator output)	125	179/167	112	154/193	125	163/151	150	167	100	142

STEAM CONSUMPTION:

897	Live steam consumption of test main turbine, water rate run, pounds per hour	68821	78352	222260	254966	344417	338004	884888	444888	471165	441888
898	Water rate of test main turbine, water rate run, lbs. of live steam per shaft horsepower hour	16.136	17.056	13.480	14.880	13.948	13.816	13.635	13.766	14.108	13.989
916	Total steam consumption, live plus auxiliary exhaust steam, of test main turbine for single unit fuel consumption runs with auxiliary exhaust admission, pounds per hour	99159	105121	246686	281842	366574	348838	--	--	--	--

FEED WATER:

900	Make up feed, gallons per hour	938	974	982	885	1080	2145	2968	1116	4880	6184
902	Make up feed, gallons per sectional mile	87.8	93.4	89.8	88.4	88.8	116	115.1	48.4	196.8	268
909	Distilled by evaporators, gallons per hour - distilled to	1000 S.T.		1000 R.T.		1000 S.T.		1000 S.T.		Secured	

*All observed data from ships' switchboard instruments unless otherwise noted.

**Values given show "pump heads" to d.c. auxiliary bus of "motor" end acting as a generator for backing action of booster set on 240 volt d.c. supply.

†12th and 13th stage shaft blading of main turbines removed, 12th and 13th stage diaphragms remaining in place.

††From soundings of all reserve feed bottoms. Due to drain collecting mains being piped to the various bottoms, all bottoms have to be sounded resulting in only approximate results being obtained for runs of short duration.

TABLE 4

MAY-JUNE 1948

SPECIAL ACCEPTANCE MACHINERY TRIALS

TABLE 8 (CONTINUED)

TABULATED TRIAL DATA

U.S.S. LEXINGTON & SARATOGA

ITEM NO.	ITEM	4800 SHP RUN	4800 SHP RUN	144 RPM RUN	144 RPM RUN	148 RPM RUN	148 RPM RUN	224 RPM RUN	224 RPM RUN	140000 SHP RUN	140000 SHP RUN
	FUEL CONSUMPTION:										
311	Gallons @ 60°F of fuel oil per hour, as fired	--	--	--	--	--	--	11288	11267	28906	28888
	- 2 hour water rate run	1407	1428	2980	3078	4128	4428	--	--	--	--
	- 2 hour fuel consumption run(Lex)	1286	--	2902	--	3946	--	--	--	--	--
	- 4 hour fuel consumption run(Sara)	--	1422	--	3093	--	4305	--	--	--	--
312	Pounds of fuel oil per hour, as fired	--	--	--	--	--	--	90999	89999	184042	185661
	- 2 hour water rate run	11209	11705	23829	24868	29012	34056	--	--	--	--
	- 2 hour fuel consumption run(Lex)	9977	--	23291	--	31485	--	--	--	--	--
	- 4 hour fuel consumption run(Sara)	--	11827	--	24365	--	33809	--	--	--	--
313	Lbs. of fuel oil per shaft hp. hour, as fired	--	--	--	--	--	--	1.331	1.326	1.311	1.314
	- 2 hour water rate run	2.339	2.547	1.348	1.395	1.310	1.386	--	--	--	--
	- 2 hour fuel consumption run(Lex)	2.136	--	1.289	--	1.254	--	--	--	--	--
	- 4 hour fuel consumption run(Sara)	--	2.567	--	1.396	--	1.348	--	--	--	--
316	Lbs. of fuel oil per nautical mile, as fired	--	--	--	--	--	--	3408	3419	5684	5749
	- 2 hour water rate run	1067	1135	1401	1456	1777	1884	--	--	--	--
	- 2 hour fuel consumption run(Lex)	950	--	1341	--	1683	--	--	--	--	--
	- 4 hour fuel consumption run(Sara)	--	1074	--	1442	--	1816	--	--	--	--
317	Lbs. of fuel oil per hour for fuel consumption runs on basis of 19800 B.t.u. per lb. of oil	9613	10704	21808	22257	30814	35079	67084	66107	177868	177715
318	Lbs. of fuel oil per hr. per sq. ft. of furnace volume for fuel consumption run on basis of 19800 B.t.u. per lb. oil	3.086	2.424	6.077	4.213	5.725	6.611	6.848	7.800	10.442	10.732
319	Lbs. of fuel oil per hr. per sq. ft. of water heating surface for fuel consumption run on basis of 19800 B.t.u. per lb. of oil	.3648	.2379	.4788	.4135	.5390	.6705	.6446	.7684	.9848	1.063
320	Lbs. of fuel oil per shaft horsepower hour for fuel consumption run, basis 19800 B.t.u. oil	2.060	2.447	1.344	1.321	1.308	1.274	1.282	1.274	1.301	1.330
321	Lbs. of fuel oil per nautical mile for fuel consumption run, basis 19800 B.t.u. oil	898	1025	1394	1394	1680	1718	3686	3276	5478	5807
322	Lbs. of fuel oil per nautical mile under contract F.p.h.and power requirements for fuel consumption run on basis of 19800 B.t.u. per lb. oil	914	1000	1340	1322	1650	1693	3547	3273	4530	5089
323	Lbs. of fuel oil per nautical mile guaranteed in contract for fuel consumption run under contract trial conditions, basis 19800 B.t.u. oil	No guar.	N.G.	1311	1311	1695	1685	3000	3005	N.G.	N.G.
324	Ratio or actual to guaranteed fuel per nautical mile under contract trial conditions	--	--	.827	.878	.967	1.005	1.114	1.089	--	--
325	B.t.u. fired per hour (multiply by 10^6) for fuel consumption run	187.5	208.7	419.4	433.5	601.8	686.5	1307.0	1279.1	3456.7	3466.4
326	B.t.u. fired per hour per cu.ft.of furnace volume for fuel consumption run	59007	47866	98008	82154	111656	129215	125636	152100	334090	339874
327	B.t.u. fired per hour per sq. ft. water heating for fuel consumption run	5564	4439	9281	8065	10611	11121	12990	14825	19804	20054
328	B.t.u. fired per shaft horsepower hour for										

886	B.t.u. fired per hour per cu.ft.of furnace volume for fuel consumption run	28007	47286	98008	82154	111656	113815	125656	154100	204090	208874
887	B.t.u. fired per hour per sq. ft. water heating for fuel consumption run	8584	4439	9581	8045	10611	11181	13970	14925	19804	20284
888	B.t.u. fired per shaft horsepower hour for fuel consumption run	40170	47717	84258	80760	83886	84065	84089	84843	85970	84880
	[redacted]										
881	Overall efficiency, fuel to shaft, all purposes	--	--	--	--	--	--	.1019	.1025	.1004	.1009
	- 2 hr. water rate run	.0816	.0857	.1005	.0988	.1026	.0998	--	--	--	--
	- 2 hr. fuel consumption run(Lex)	.0854	--	.1060	--	.1062	--	--	--	--	--
	- 4 hr. fuel consumption run(Sara)	--	.0854	--	.0988	--	.1068				

[redacted]

/Contract guarantee values are for machinery plant with 13 stage turbines as originally installed. Actual values for this set of trials were obtained with the 12th and 13th stage wheel binding of turbines removed with diaphragms remaining in place.

TABLE 6

PARAMETER PERFORMANCE DATA

SS KEY HIGHBANKS NEK - SAN DIEGO, CALIFORNIA, TO HONOLULU, TERRITORY OF HAWAII - JUNE 9-12, 1939 - U. S. S. ENTERPRISE

ITEM NO.	ITEM	1st 24 HOURS	2nd 24 HOURS	3rd 24 HOURS	CONTINUE NEK
1	Speed of vessel in knots				
	- from propeller r.p.m.(Bby,1920,94in.)	30.17	31.12	31.96	30.89
	- true, from distance made good	29.85	30.67	31.00	30.44
	- estimated current, knots	0	+.28	+1.28	+.30
1a	Miles steamed				
	- from propeller r.p.m.(Bby,1920,94in.)	724	737	768	2242.5
	- true, from astronomical observations	704	736	768	2208
2	Duration of run, hours	24	24	24	72.6
3	Time commencing run	--	--	--	Zone +8,1301½
4	Date, June 1939	9-10th	10-11th	11-12th	June 9-12,1939
5	Locality	Great circle route from San Pedro, Calif., to Honolulu, T.H.			
6	Days out of dock	46	49	50	49
7	Displacement	--	--	--	Service
9	Sea, condition and relative bearing	0-2-2-1,0-180°	1,180°	1,180°	1st day 2,0-180° Remainder 1,180°
10	Relative wind - velocity, knots	24.4	17.4	14.9	22.2
	- bearing, degrees	30	0	0	10
11	Barometer, inches Hg.	--	--	--	30
12	Outside air temperature, degrees F.	61.4	68.1	73.6	68.5
13	Temperature of sea, degrees F.	62.8	70.0	73.7	69.7
21	Number of boilers in use	--	--	--	18
22	Number of main turbo-generators in use (12th & 13th stage wheel buckets removed)	--	--	--	4
23	Number of main motors in use	--	--	--	2
24	Main motor pole combination	--	--	--	22
25	Number of main motor vent fans in use	--	--	--	8
26	Number of forced draft blowers in use	18	18	20	20
30	Number of fuel oil burners in use	96	103	104	101
	- size of sprayer plates	2008	2008	2008	2008
31	Number of main feed pumps in use	--	--	--	6
34	Number of main circulating pumps in use	--	--	--	8
39	Number of main turbo-gen.airfoil cooler pumps in use	--	--	--	4
47	Number of main motor rm. airfoil cooler pumps in use	--	--	--	2

83	Number of main feed pumps in use	--	--	--	3
84	Number of main circulating pumps in use	--	--	--	3
86	Number of main turbo-gen.airfoil cooler pumps in use	--	--	--	4
87	Number of main motor gen. airfoil cooler pumps in use	--	--	--	2
89	Number of quadruple effect distilling plant units	Both units in operation for a total of 45 hours			
90	Refrigerating plant	After plant in operation for entire run; forward plant for 12 hrs.			
90a	Heating system	--	--	--	Secured
90b	Other auxiliary machinery in use	Standard practice			
96	Disposition of auxiliary exhaust steam	Feed heaters, distilling plant, & 8th stage of main turbines, and turbine glands.			
97	Disposition of drains	Standard practice			
81	Specific gravity of fuel oil, degrees A.P.I.	--	--	--	16.2
82	Viscosity of fuel oil - - - - @ deg. Engler at	--	--	--	309.7
88	Heat content of fuel oil, B.t.u. per lb., as fired(est)	--	--	--	18900
91	Steam pressure at boilers, lbs.per sq. in., gage	280	281	276	280
98	Steam pressure in engine rooms, lbs.per sq.in., G.	270	271	272	271
99	Main turbine exhaust pressure, inches Hg.vacuum	27.5	27.0	26.7	27.0
101	Auxiliary exhaust pressure in engine rooms	10	14	10	11.3
128	Superheat at boilers, degrees F.	57	42	38	45
142	Temperature feed water from feed heaters, deg.F.	237	227	226	230
195	Warmest main propuls.machy.bearing temp., deg. F.	142	148	147	149
199	Warmest machinery or boiler compartment temperature 6 ft. above floor plates, deg. F.	110	115	122	122

TABLE A (CONTINUED)

TABULATED PERFORMANCE DATA

NO KNOT ENDURANCE RUN - SAN PEDRO, CALIFORNIA, TO HONOLULU, TERRITORY OF HAWAII - JUNE 9-16, 1938 - U. S. S. LEXINGTON

ITEM NO.	ITEM	1st 24 HOURS	2nd 24 HOURS	3rd 24 HOURS	CONTRACT RUN
#31	Main generator line volts	4129	4154	4206	4146
#38	Main generator line amperes	5644	5752	5754	5677
#36	Main generator line power factors	.86	.87	.87	.867
#37	Main generator line indicated kilowatts,total	88776	98676	98998	98816
#40	Main gen.line kw.from $\sqrt{3}$ x E x I x cos θ,total	87450	96100	95850	92000
#49	Main generator field volts	142	139	142	141
#51	Main generator field amperes	567	576	582	582
#41	Speed of vessel in knots corresponding to propeller r.p.m.,from stds.of May 1938	30.19	31.12	31.36	30.89
#41a	Ratio miles steamed from r.p.m.to miles made good	1.028	1.014	.979	1.007
	- ditto, corrected for estimated current	1.028	1.028	1.010	1.025
#42	Total shaft horsepower - from control board instms	109471	116792	119634	115436
	- from propeller r.p.m.,May 1938 stds.	104450	114698	118804	112647
#42a	Ratio S.Hp.by instruments to S.Hp.by r.p.m.	1.0478	1.019	1.009	1.026
#47	R.p.m. of propellers, all shafts average	261.3	270.4	273.2	260.48
#91	Total d.c.auxiliary power for all purposes, total output of 750 kw. turbo-generators, kw.	1295	1079	1275	1195
#01	Make-up feed consumption - gallons per hour	848	1561	2160	1614
	- tons per hour	3.19	5.81	8.04	6.00
	- gallons per day	20522	37464	51840	38754
	- tons per day	76.12	139.4	193.0	144
#02	Make-up feed consumption - gals.per naut.mile(rpm)	27.97	50.16	68.9	52.2
#11	Fuel consumption in gallons of fuel oil at 60°F per hour as fired	17184	18268	19119	18290
#12	Pounds of fuel oil per hour,as fired	124606	140154	138410	132802
#12a	Tons of fuel oil per hour, as fired	60.96	66.14	68.04	66.09
#18	Lbs. of fuel per hr.per sq.ft.furnace vol., as fired	8.6	9.38	9.60	9.19
#24	Lbs. of fuel oil per hour per sq. ft. of water heating surface, as fired	.61	.679	.704	.664
#28	Lbs. of fuel oil per shaft hp. hour, as fired	1.247	1.270	1.278	1.268
#61	Overall efficiency, fuel to shaft, all purposes, %	10.09	10.66	10.61	10.78

NOTE:

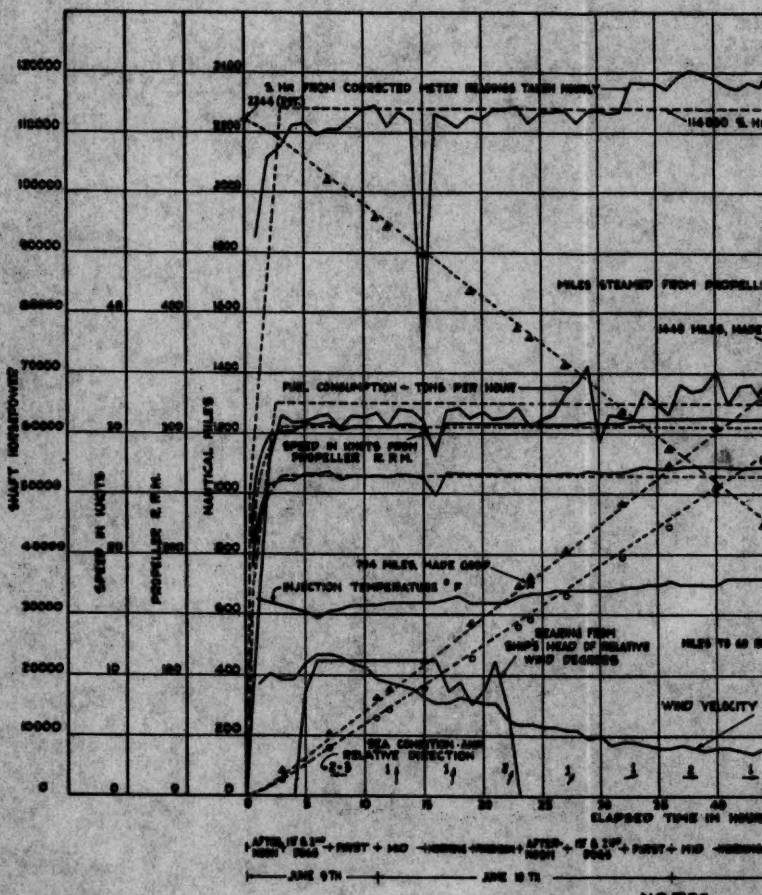
...the 16th and 18th wheel buckets of all main turbines removed and the 18th and 19th

850	Lbs. of fuel oil per shaft Hp. hour, as fired	1.367	1.370	1.378	1.366
851	Overall efficiency, fuel to shaft, all purposes, %	10.09	10.46	10.41	10.78

[redacted]:

This run was made with the 12th and 13th stage wheel buckets of all main turbines removed and the 12th and 13th stage diaphragms left in place.

[redacted] 6
[redacted] 1



LEGEND

BROKEN LINES SHOW ESTIMATED PERFORMANCE.

FULL LINES AND SPOTTED POINTS SHOW ACTUAL PERFORMANCE

NOTES

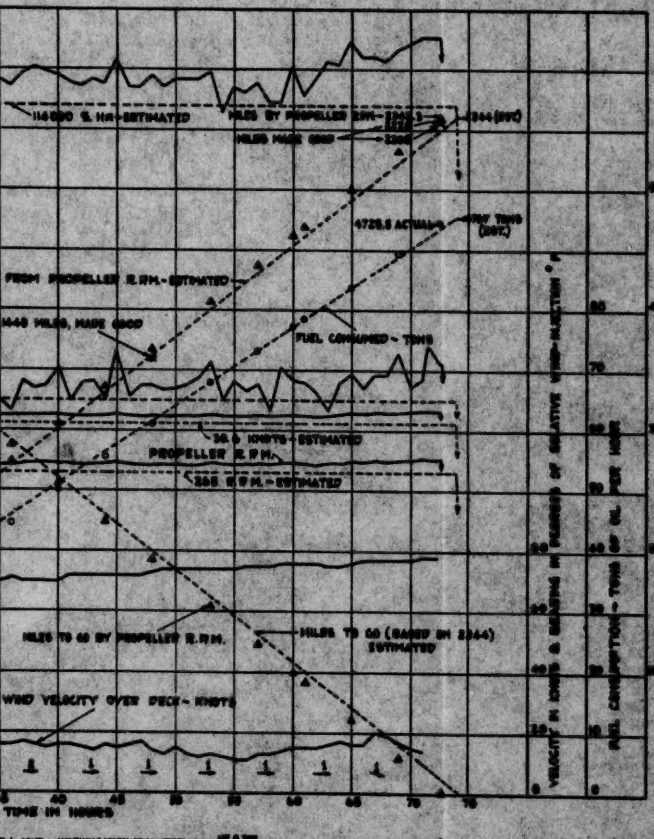
LEFT SAN PEDRO ANCHORAGE - 12" SWELL 1341-20 JUNE 4TH, 1935, ZONE + 8 TIME

DIAMOND HEAD LIGHT ABEAM (BEARING NORTH) 1107-20 JUNE 10TH, 1935, ZONE + 10 TIME

CURVATURE: 870 MINS. LOSS TERMINAL ON NO. 1A MAIN MOTOR VENT FAN FUSE BLACK DESIRING TEMPORARY SHUTTING OFF OF MAIN POWER TO THROW VENT FAN SWITCHES.

TURBINES OPERATING UNTRUST 12TH & 13TH STEAM

FIGURE 8.


FROM PROPELLER R.P.M.-ESTIMATED
FUEL CONSUMED - TONS
PROPELLER R.P.M.
MILES TO GO BY PROPELLER R.P.M.
MILES TO GO (BASED ON 2344) ESTIMATED
WIND VELOCITY OVER DECK - KNOTS
TIME IN HOURS
FUEL CONSUMED - TONS OF 2240 POUNDS

JUNE 11TH — JUNE 12TH

2 TKS

ESTIMATED MILES THROUGH WATER BY PROPELLER R.P.M. 2344.0 IN 74.5 HOURS
ACTUAL 2342.2 IN 72.0 HOURS(GREAT CIRCLE COURSE)

ESTIMATED CURRENT AVERAGE OF + .8 KNOT FOR 74.5 HOURS (27 MILES OR LESS HOURS AT 30 KNOTS)-USED AS RESERVE MARGIN- ACTUAL AVERAGE CURRENT PREVAILY +.8 KNOT FOR 72.0 HOURS.

2nd & 10th STAGE RATOR BLADING

GURE 8.

30 KNOT ENDURANCE RUN PERFORMANCE

SAN PEDRO, CALIF. TO HONOLULU, T. H.
JUNE 9TH TO 12TH 1939
U.S.S. LEXINGTON

TABLE 7

TABULATED TRIAL DATA

FINAL MACHINERY TRIALS

ITEM NO.	ITEM	4 UNITS AT 46500 SHP		4 UNITS AT 144 RPM		4 UNITS AT 146 RPM		U. S. S. YORKTOWN & ENTERPRISE: 4 UNITS AT 226 RPM		4 UNITS AT 272 RPM		4 UNITS AT 180000 SHP	
		LEX	PARK	LEX	PARK	LEX	PARK	LEX	PARK	LEX	PARK	LEX	PARK
	GENERAL DATA:												
1	Speed, knots (from curves Nov-Dec 1940 Trials)	10.88	10.68	16.91	16.90	19.10	18.90	26.63	26.68	31.66	31.67	34.24	33.91
2	Duration, hours- water rate	2	2	2	2	2	2	—	—	—	—	—	—
	- fuel expense	4	4	4	4	4	4	—	—	—	—	—	—
	- T.H. & F.C.	—	—	—	—	—	—	4	4	4	4	4	4
3	Time commencing run	0615	0600	0700	1000	0730	0600	0930	1130	0800	0800	0800	0800
4	Date, 1940	Nov 17th	Dec 16th	Nov 19th	Dec 11th	Nov 12th	Dec 12th	Nov 10th	Dec 13th	Nov 14th	Dec 17th	Nov 16th	Dec 15th
5	Locality	So.Calif Coast	So.Calif Coast	So.Calif Coast	Gen.Cal. Coast	So.Calif Coast	So.Calif Coast	So.Calif Coast	So.Calif Coast	So.Calif Coast	So.Calif Coast	So.Calif Coast	So.Calif Coast
6	Days out of dock	9	8	8	1	4	2	2	3	6	7	7	8
7	Displacement, tons	36925	36975	36027	36989	36950	36996	36800	36964	36851	36849	36866	36926
8	Trim by the stern, at anchor aprx	4'-6"	4'-0"	3'-8"	2'-6"	4'-6"	3'-7"	4'-3"	3'-7"	4'-8"	3'-7"	4'-8"	3'-7"
9	Sea - state, Beaufort scale	0 & 1	0 to 1	3	4	3	0 to 1	3	4	4	4	2	0 to 1
	- relative bearing, deg.	0 & 180	Variable	45 & 225	Var.	90 & 225	Var.	110	Var.	30 & 200	0 to 1 Var.	225 & 45	Var.
10	Relative wind -velocity, knots	9	7 to 15	42 & 9	11 to 22	25 &15	12 to 25	25	15 to 41	27	25 to 39	22	37 to 41
	-bearing, degrees	0	Variable	15 & 230	Var.	25 & 240	Var.	350	Var.	340	Var.	340	Var.
11	Barometer, inches Hg.	30.08	30.13	29.90	30.03	2989	30.07	29.99	29.98	29.87	30.04	29.99	30.18
12	Outside air temperature, deg.F.	60	58	61	58	62	56	61	57	62	66	61	54
13	Temperature of sea, deg. F.	60	58	61	55	62	58	61	59	59	58	60	58
	MACHINERY SET-UP:												
21	Number of boilers-W.Rate/Fuel Con.	5/5	4/3.5	4/4	5/5	5/5	5/5	10	10	16	16.5	16	16
22	-Boilers number (9-11)	(B)	(A)	(D)	(A)	(D)	(A)	(CAD)	(AEC)	(All)	(All)	(All)	(All)
23	Number Of main turbo-generators	1	1	1	1	1	1	2	2	4	4	4	4
24	Main motor pole combination	4	4	4	4	4	4	4	4	4	4	2	2
25	No.main motor vent fans, total of 16 fans(8 125 hp & 8 250 hp motor)	44	44	44	44	44	44	22	22	8	8	8	8
26	No.forced draft blowers- V.R./F.C.S./S.	4	4	4	4	4	4	4	4	8	8	8	8
	Total hand man. valves open	6/6	4/3.5	4/4	5/5	5/5	5/5	10	11	17.4	20.6	22	22
27	No.fuel oil booster pumps	2	6/6.7	6/8	12.5/10	10/10	10/10.8	22	48.7	56.1	120.0	64	128
28	No.fuel oil service pumps	2	2	2	2	2	2	4	2	2	2	2	2
29	No.fuel oil h eters	2/24	2	2	2	2	2	2	2	2	2	4	4
			4/3.5										

	15 Same(@ 125 hp & 4 200 hp mtrs)	4	4	4	4	4	4						
26	No.forced draft blowers- F.R./F.O.-H/PC		4/3.8	4/4	4/5	5/5	5/5	10	11	17.4	20.4	32	32
	Total hand man. valves open	8/8%	8/6.7	8/8	12.8/10	10/10	10/10.8	22	42.7	58.1	120.0	64	128
27	No.fuel oil booster pumps	2	2	2	2	2	2	4	2	2	2	4	2
28	No.fuel oil service pumps	2	2	2	2	2	2	3	3	2	4	6	6
29	No.fuel oil h eters	8/2%	4/3.3	4/4	8/8	8/8	6/6	10	11	16	18.3	16	16
30	Total No.fuel oil burners-HH/PC	7/8%	17/15.8	20/17	34/26	29/23	18/18 35/30,3	47	55	80	91.4	122	127.7
30a	Size of sprayer plates	2000	2800	2800	2800	2800	2800	2000	2000	2000	2000	2000	2000
31	No. main feed pumps	2	2	2	2	2	2	3	3	4	5	8	7
	Total hand man.valves open	1	0	1	0	1	0	3	0	4	1	4	1
32	No. main feed heaters	2	2	2	2	2	2	2	2	4	4	4	4
33	No. make-up feed pumps,igternit.serv	1	1	1	1	1	1	1	1	1	1	1	1
34	No. main circulating pumps	1	2	1	2	1	2	2	4	4	8	8	8
	Total hand man.valves open	0	0	0	0	0	0	0	2	0	1	0	4
35	No. main air pump units	1	2	1	2	1	2	2	4	4	8	8	8
	Total No. 2-stage ejectors	1	4	1	4	1	4	3	8	6	16	16	16
36	No. main condensate pumps	1	1	1	1	1	1	2	2	4	4	7	5
	Total hand man.valves open-TS/PC	0	0	0/1	0	1	0	0	0	2	0	2	2
37	No. main turb. forced lub.oil pumps	1	1	1	1	1	1	2	2	4	4	4	4
	Total hand man.valves open	0	0	0	0	0	0	0	0	0	0	0	0
38	No. main turb. lub. oil coolers	3	3	3	3	3	3	6	6	12	12	12	12
39	No. main turb,airdoil c.oler & hydraulic vacuum pump supply pumps	1	1	1	1	1	1	2	2	4	4	4	4
40	No. main generator air cooler banks	16	16	16	16	16	16	32	32	64	64	64	64
41	No. 750 kw. d.c. turbo-generators	2	2	2	2	2	2	2	4	2	4	2	2
42	No. 750 kw.set condens.plants-HH/PC	2	2	2/2	2	4/2	2	2	4	4	4	2.7	4
43	No. booster motor-generators	1	1	1	1	1	1	2	2	4	4	4	4
44	No. lighting motor-generators	2	2	2	2	2	2	2	2	2	2	2	2
45	No. motor room lub.oil pumps(HB)	4	4	4	4	4	4	4	4	4	4	4	4
46	No. motor room lub.oil coolers	0	0	4	4	4	4	4	4	4	4	4	4
47	No. motor room airdoil cooler pumps	2	4	2	4	2	4	2	4	2	4	4	4
48	No. main motor air cooler banks	24	24	24	24	24	24	24	24	48	48	48	48
49	No. gundr.offset distil.plant units	1	1.4	1	1	1	1.8	1	1.4	1	1	0	2
50	Refrigerating plant set-up --fwd.	0	0	0	0	0	0	0	0	0	0	0	0
	(Lists of Compliance) --aft	0	0	0	1	0	1	0	1	0	1	0	1

50a Heating system — about 12% capacity for Lexington; about 20% capacity for Saratoga

*Numbers 6 and 15 boilers cut out at 1035

Note: Values listed are for total runs except where otherwise noted and where separated by / ; values to left of / are for water rate runs to right of / for fuel consumption runs.

TABLE 1 (CONTINUED)

NOV. - DEC. 1928

TABULATED TRIAL DATA

FINAL MACHINERY TRIALS

U. S. S. LEXINGTON & SARATOGA

ITEM NO.	ITEM	[illegible] LEX	SARA	[illegible] LEX	SARA	[illegible] LEX	SARA	[illegible] LEX	SARA	[illegible] LEX	SARA	[illegible] LEX	SARA
	MACHINERY SET-UP (Continued):												
51	No. steam fire & bilge pumps - on firemain	0	1	0	1	0	1	0	1	0	1	0	1
	on bilges, water rate/fuel on.	0/[illegible]	0	1/0	0	1/0	[illegible]	[illegible]	[illegible]	1	1	1	1
52	No. elec. F&B pumps on firemain (MD)	1	0	1	0	1	0	1	0	1	0	1	0
53	No. CAR flushing pumps (MD)	1	1	1	1	1	1	1	1	1	1	1	1
54	No. ships service air comp. (MD) (Intermittent service)	1	2	1	2	1	2	1	2	1	2	1	2
55	Steering gear set-up	Std.	Std.	Std.	Std.	Std.	Std.	Std.	Std.	Std.	Std.	Std.	Std.
56	Disposition of aux. exhaust steam:- To main turbines - WR / FC	No/Yes	No/Yes	No/Yes	No/Yes	No/Yes	No/Yes	C	C	A.B.C	B.C.D	A. B. C	B.C. D
	-stage admission - WR / FC.	--/8th	--/8th	--/8th	--/8th	---/8th	--/10th	8th	10th	[illegible]	10th	10.10.10	10th
	No. feed heaters admitted to	2	2	2	2	2	2	2	2	4	4	4	4
	No. main exh. trunks - WR / FC	0	0/1	0	0/1	0	0/1	0	1	0	1	0	2
	No. dyn. condensers - WR / FC	1/0	3/0	1/0	3/0	2/0	3/0	0	0	0	0	0	0
	No. distil. plant units - WR/FC	1	2/1.4	1	1	1	2/1.8	1	1.4	1	1	0	2
57	Disposition of drains:- High pressure to -	Fd.Htr.	F.H.&F.T.	F.H.	F.H.&F.T.	F.H.	F.H.&F.T.	F.H.	F.H.&F.T.	F.H.	F.H.&F.T.	F.H.	F.H.&F.T.
	Low pressure to -	Fd.Tnk.	F.T.	F.T.	F.T.	F.T.	F.T.	F.T.	F.T.	F.T.	F.T.	F.T.	F.T.
	Feed heater to -	Fd.Tnk.	F.T.	F.T.	F.T.	F.T.	F.T.	F.T.	F.T.	F.T.	F.T.	F.T.	F.T.
	1st effect evaporator to -	Dyn.Cond.	D.C.	D.C.	D.C.	D.C.	D.C.	D.C.	D.C.	D.C.	D.C.	None	D.C.
	DIMENSIONAL DATA:												
61	Total connected boiler water heating surface, sq.ft. WR / FC	33750 / [illegible]	45000 / 37125	45000 / 45000	56250 / [illegible]	56250 / 56250	67500 / 67500	112500	123750	180000	172125	180000	180000
62	Total con. blr. suphtg. surf., sq.ft.	2201 / 1793	4364 / 3800	3188 / 3188	5455 / 5455	3985 / 3985	6546 / 6546	7970	12001	12752	18698	12752	17456
63	Ttl. con. blr. fur. vol., cu.ft.	3177 / [illegible]	4418 / 3843	4234 / 4234	5520 / 5520	5296 / 3985	6624 / [illegible]	10590	12144	18944	18091	18944	17884
64	Ttl. [illegible] surf., sq.ft.	32368	30094	32368	30094	32368	30094	64736	60188	129472	120376	129472	120376
65	Ttl. con. dyn. cond. surf., sq.ft.	4028	6042	6042 / 4028	6042	8056 / 4028	6042	4028	8056	8056	8056	7452	8084
66	Ttl. con. fd. htg. surf., sq.ft.	2340	2340	2340	2340	2340	2340	3510	2340	4680	4680	4680	4680
66a	Ttl. con. f.o. htr. surf., sq.ft.	[illegible]		404		808		1736		2778		2778	
67	Ttl. con. evap. htg. surf., sq.ft.	1640	2296	1640	1640	1640	2952	1640	2296	1640	1640	0	3280
68	Ttl. con. mn. gen/air coolg. surf., sq.ft.	23374	23374	23374	23374	23374	23374	46748	46748	93496	93496	93496	93496
69	Ttl. con. mn. mtr/air coolg. surf., sq.ft.	28580	28580	28580	28580	28580	28580	28320	28320	56640	56640	56640	56640
70	Ttl. con. main turbo-generator oil cooler surface, sq.ft.	900	900	900	900	900	900	1800	1800	3600	3600	3600	3600

68	Ftl.com.sm.gen.Ahlg.surf.sq.ft.	25974	25974	25974	25974	25974	25974	46748	46748	93496	93496	93496	93496
69	Ftl.com.sm.evp.Ahlg.surf.sq.ft.	26580	26580	26580	26580	26580	26580	26580	26580	54640	54640	54640	54640
70	Ftl.com.main turbo-generator oil cooler surface, sq.ft.	900	900	900	900	900	900	1800	1800	3600	3600	3600	3600
71	Ftl.com.sm.motor & thrust block oil cooler surf.sq.ft.	0	0	1800	1800	1800	1800	1200	1200	1800	1800	1800	1800
72	Ftl.com.750 kw.turbo-gen. oil cooler surface, sq.ft.	220	220	220	220	220	220	220	440	220	440	220	440
73	Ftl.com.sm.turb.capacity.rated hp	47168	47168	47168	47168	47168	47168	94336	94336	188672	188672	188672	188672
74	Ftl.com.sm.mtr.capac.,rated hp.	*88000	*88000	*88000	*88000	*88000	*88000	90000	90000	180000	180000	180000	180000
75	Ftl.com.main motor vent fan motor capacity,rated hp.	500	500	500	500	500	500	500	500	1000	1000	1000	1000
76	Ftl.com.750 kw.turbo-gen. capacity, rated kilowatts	1500	2250	1500	2250	1500	2250	1500	3000	2250	3000	2250	3000
77	Mean pitch of propellers, all shafts average	Lexington, 15 feet 2.15 inches; Saratoga, 15 feet 2.50 inches.											
78	Mean pitch of individual propellers, Nov.-Dec. 1928 dry docking												
	No. 1 shaft	Lexington	15 ft. 0.98 in.	15 ft. 1.80 in.		15 ft. 2.13 in.		Saratoga	15 ft. 2.84 in.	15 ft. 2.835 in.		15 ft.2.89	
	No. 2 shaft		15 ft. 2.95 in.						15 ft. 2.77 in.				
	No. 3 shaft		15 ft. 1.68 in.	15 ft. 2.46 in.					15 ft. 2.92 in.	15 ft. 2.875 in.			
	No. 4 shaft		15 ft. 3.04 in.						15 ft. 2.40 in.				
	FUEL OIL CHARACTERISTICS:												
81	Specific gravity, degrees A.P.I.	16.4	16.5	16.1	16.2	16.3	16.2	16.4	16.2	16.4	16.5	16.5	16.5
82	Pounds per gallon @ 60°F	7.96	7.96	7.98	7.97	7.96	7.97	7.96	7.97	7.96	7.96	7.96	7.96
83	Viscosity: 4 deg.Engler at	218°F	208°F	216°F	196°F	214°F	196°F	208°F	196°F	211°F	208°F	218°F	208°F
84	Water & sediment, per cent	0.4	0.5	0.5	0.4	0.45	0.40	0.40	0.35	0.40	0.35	0.30	0.40
85	Flash point, deg. Fahr.	220	230	230	230	225	235	225	231	220	228	225	230
86	Source	Calif.	Calif.	Calif.	Calif.	Calif.	Calif.	Calif.	Calif.	Calif.	Calif.	Calif.	Calif.
87	Heat content,Btu per lb,water free	18906	18900	18894	18860	18902	18888	18906	18880	18900	18877	18900	18888
88	Heat content, Btu. per pound as fired - - - - - - -	18826	18800	18795	18770	18826	18774	18826	18745	18821	18767	18830	18741

*44 pole combination rating

TABLE I (CONTINUED)

NOV. - DEC. 1928 — TABULATED TRIAL DATA

FINAL MACHINERY TRIALS — U. S. S. LEXINGTON & SARATOGA

ITEM NO.	ITEM	4 HRS AT 4500 SHP LEX	SARA	4 HRS AT 144 RPM LEX	SARA	4 HRS AT 148 RPM LEX	SARA	4 HRS AT 268 RPM LEX	SARA	4 HRS AT 275 RPM LEX	SARA	4 HRS AT 180000 SHP LEX	SARA
	PRESSURES:												
91	Steam at boiler, lb. per sq.in.Gg. -water rate run/fuel con.run	273/272	276/279	281/276	281/278	280/279	279/278	268	288	279	291	298	299
92	Steam in engine rooms- -forward, lbs.gage, WR/FC	274/273	278/275	278/275	275/272	276/278	273/270	281	288	275	275	282	275
	-after, lbs.gage, WR/FC	273/273	264/262	279/274	270/268	278/278	270/278	278	282	276	268	285	281
93	Steam at test main turbine throttle, G.E.Co. gage as read, lbs.gage, WR/FC	268/260	276/278	275/270	278/271	268/268	269/267	265	269	268	276	271	278
94	Steam ahead main turb, lbs.G, WR/FC	270/270	275/276	273/270	277/268	269/271	266/264	268	263	263	271	261	266
	-"A" turbine	—	275/276	—	277/268	—	266/264	—	264	265	272	262	265
	-"B" turbine	—	—	—	—	—	—	—	—	260	274	256	258
	-"C" turbine	—	—	—	—	—	—	259	263	261	270	259	264
	-"D" turbine	270/270	—	273/270	—	269/271	—	265	—	263	270	265	261
95	1st stage main turb, lbs.G, WR/FC	14/9	12/9	72/69	70/66	102/102	101/99	147	153	120	118	197	188
	-"A" turbine	—	12/9	—	70/66	—	101/99	—	154	117	116	193	185
	-"B" turbine	—	—	—	—	—	—	—	—	126	124	205	195
	-"C" turbine	—	—	—	—	—	—	149	152	114	116	196	187
	-"D" turbine	14/9	—	72/69	—	102/102	—	146	—	121	116	195	185
96	8th stg.adm.belt, lbs.G, or in.Hg.vac. (See item 86) -water rate run	26"	18.8"	9.5"	3.0	2.2"	3.2						
	-fuel consump. run	11.7"	0.1"	3.8"	4.6	.44"	2.4						
	-combined WR & fuel con.run: "A" turbine (no aux.exh.admission for Saratoga)							—	5.0	8.2	0.0	9	9.0
	"B" turbine							—	—	3.9	5.6	10.2	6.2
	"C" turbine							7.1	6.4	2.8	1	11.2	10.8
	"D" turbine (no aux.exh.admission for Lexington)							3.7	—	1.6	2	10.6	11.3
97	10th stg.ad.belt, lbs.G, or in.Hg.vac. (See item 86) -water rate run	30"	25"	24.3"	18.4"	20.7"	8.8"						
	-fuel consump. run	29.9"	24.4"	23.9"	13.2"	20.2"	8.3"						
	-combined WR & FC run: "A" turbine (no aux.exh.adm.for Saratoga)							—	9.8"	11"	15"	1.6	9.4"
	"B" turbine							—	—	14.2"	18.4"	3.0	4.0"
	"C" turbine							11.9"	7.1"	9.0"	10.6"	4.1	6.2"
	"D" turbine (no aux.exh.adm.for Lexington)							13.0"	—	10.1"	18.0"	10.7"	2.8"
98	Main turb.exh., in.Hg.vac.(Og.bd.) -water rate run	29.2	29.0	28.4	29.0	28.2	28.6						
	-fuel consump. run	28.7	29.0	28.5	28.5	28.1	29.0						
	-combined WR & FC run							27.3	28.6	27.7	29.1	27.0	28.5
99	Test main turbine exh., G.E.Co. abs.press.gage, in.Hg.abs. -WR	1.09	1.28	1.68	1.46	1.905	1.84						
	-FC	1.09	1.38	1.81	1.39	1.905	1.99						
	WR&FC							2.83	2.5	2.848	1.97	3.168	2.78

	-Fuel economy, run		28.7	29.0	28.5	28.5	28.1	29.0					
	-combined ME & FC run								27.8	28.5	27.7	28.1	27.0	28.5
99	Test main turbine sub.,G.E.Co.													
	abs.press.gage,in.Hg.abs.-ME	1.00	1.38	1.48	1.46	1.905	1.84							
	-FC	1.39	1.38	1.81	1.89	1.905	1.99							
	Mean								2.85	2.5	2.845	1.97	2.165	2.75
100	Dynamo condenser vac.,in.Hg/deg.Rd	29.4	29.4	29.8	27.8	27.7	27.9		28.9	27.8	29.6	27.7	28.1	27.9
101	Aux.exhaust steam,at aux.turb., lbs.per sq.in.dg.- ME / FC	11/10	10/12	11/10	12/13	10/11	12/12.5		11.5	12.8	10.0	10.6	9.0	10.9
102	Forced draft blower turb.steam boil,lbs.gage - ME / FC	188/182	100/97	188/181	193/184	168/188	174/204		208	208	209	184	194	194
103	F.D.blower exhaust,lbs.dg.-ME/FC	13/12	--	11.7/11	--	12.4/13.4			12.1	--	11.6	--	11	--
104	Fireroom air, in.water- ME/FC	2.8/2.9	1.4/1.1	2.3/2.1	1.8/1.8	2.4/2.3	2.0/1.8		2.9	2.3	3.8	4.8	6.3	6.5
105	Fuel oil pump disch,lbs.dg.	275	280	300	249	300	246		308	288	315	300	311	288
106	Fuel oil @ burner manifold,lbs.dg.	290	240	289	238/229	296	216/249		290	269	295	221	290	249
106a	Fuel oil htr.steam press,lbs.G.	16/22		54/70		54/72			44		41		48	
107	Main feed pump disch.,lbs.gage	380	371	381	382	381	371		339	304	333	309	288	288
108	Main feed at boilers, lbs.gage	349	334	330	322/265	341	254		338	301	327	299	247	262
109	Mn.turb.lub.oil pump disch..lbs.dg.	82	81	84	90	86	88		85	81	88	81	89	79
110	Mn.str.mn.lub.oil to bearg.,lbs.dg.	6	8	10	4½	11	5½		11	5½	10	5½	9	8
111	Lub.oil to main turb.gov. lbs.dg.	81	80	85	90	86	88		86	80	86	81	89	78
112	Lub.oil to mn.turbo-gen.bearing supply line, lbs. dg.	34	30	30	35	27	35		34	27	27	27	34	30
113	Lub.oil at mn.tur-gen.bearg,lbs.dg.	12	15	12.5	15	12.5	15		15	13.5	13	15	13.5	15
114	Main cond.circ.pump disch..,lbs.dg.	17	10	17.5	12.5	16.5	13		17.5	21	18	17.5	18	18.7
115	Mn.cond.condensate pump disc,lbs.dg.	8	7.5	13	14.5	12.0	15		10.5	8	8	10.5	8.5	9.5
116	Dyn.cond.circ. pump disch.,lbs.dg.	11.5	**9	11.0	**8	11.9	**8		12.2	**8	11.0	**8.5	11.5	**8
117	Dyn.condensate pump disch.,lbs.dg.	8	--	7.5	--	7.9	--		7.4	--	7.0	--	6.5	--
118	Mn.air pump steam supply, lbs.dg	160	150	150	150	150	150		154	167	154	172	155	171
119	Mn.air pump suction vac.,"Hg	29	29	29.7	29.8	30	29		29.5	29.2	29.3	29.2	29.0	29.4
120	Dyn.air pump steam supl.,lbs.dg.	165	175	160	168	190	159		175	165	175	164	170	163
121	Dyn.air pump suction vac.,in.Hg.	29.7	28	29.6	29.5	29.4	29.3		29.5	29.5	29.5	27.1	29.6	29.4
122	Firemain pump disch.,lbs.gage	65	60	58	61	90	69		60	52	65	60	65	68
123	Main feed heater inlet, lbs.gage	348	374	362	378	348	364		348	305	347	375	348	309
124	Main feed heater outlet,lbs.gage	342	377	362	370	348	364		348	305	347	375	355	309
125	Main feed htr. shell,lbs.dg	13.5/12.1	12.5/13	12.6/11.6	12/14	12.5/14	12.5/13		11.5	11.5	11.5	12	10.5	12
126	Mn.Gen.Air&Oil cooler pump,lbs.dg.		15		18		17			12		14		16.5
127	Mn.Gen.Hyd.Vac.Pump Sup.pump,lbs.G.	35		40		40			40		38		36	

*Lexington gages read pressure at dynamo flat cross connection in each engine room; Saratoga gages read pressure at forward cross connection of forward engine room and at after cross connection of after engine room.

**@ 700 kw. turbo-generator oil coolers.

TABLE 7 (CONTINUED)

NOV. - DEC. 1928

TABULATED TRIAL DATA

FINAL MACHINERY TRIALS

U. S. S. LEXINGTON & SARATOGA

ITEM NO.	ITEM	4 HRS AT [illegible] LEX	SARA	4 HRS AT 144 RPM LEX	SARA	4 HRS AT 162 RPM LEX	SARA	4 HRS AT [illegible] RPM LEX	SARA	4 HRS AT 272 RPM LEX	SARA	4 HRS AT 180000 SHP LEX	SARA
	TEMPERATURES:												
131	Steam from boiler suph.,°F-WR/FC	435/435	461/459	448/444	476/476	453/448	474/474	447	485	450	494	472	501
132	Superheat at boiler, °F -WR/FC	19/20	46/43	38/29	60/62	37/32	58/59	30	66	35	65	32	79
133	Steam @ test turb.throttle, °F, - water rate / fuel con.	428/431	468/452	445/446	473/470	448/447	469/471	440	487	455	482	474	498
134	Supht.@ test turb.thr.,°F-WR/FC	16/19	31/29	31/33	58/57	36/30	56/59	37	74	45	67	61	75
135	Mn.turb.steam chest,°F - WR/FC	430/435	480/485	449/447	475/474	450/446	474/473	441	486	448	489	467	496
136	Supht.@ mn.turb.st.ch.,°F-WR/FC	17/20	36/40	35/34	60/62	37/33	62/62	31	75	37	76	57	85
137	Main turb.exhaust, °F - WR/FC	82/81	88/88	92/92.4	90/94	100/100	98/100	111	109	108	106	116	112
137a	Test mn.turb.exh., °F - WR/FC	82/81	88/88	92/92.4	90/94	100/100	98/100	112		104.5		117	
138	Evaporator 1st effect tube nest header(aux.exh), °F	235	239	222	--	222	235	220	236	270	240	Secured	240
	- superheat, °F	17	16	5	--	5	21	Wet	26	58	18	--	18
139	Fuel oil booster pmp.suct., °F	88/94	82	78/77	68	80/87	69	70	89	84	83	94	86
140	Fuel oil to burners, °F	195/193	176	193/196	161	192/196	161	196	161	198	160	197	178
141	Main feed heater inlet, °F	112	135	100	122	100	123	108	123	104	117	104	117
142	Main feed heater outlet, °F	229	231	222	232	221	211	219	212	208	212	204	207
143	Main injection, °F	60	67.5	61	53	62	58	61	59	59	56	60	58
144	Main cond.ovbd.discharge, °F	65	65	73	64	78	71	85	75	78	71	82	75.6
145	Mn.cond.circ.water rise, °F	5	7.5	12	11	16	13	24	16	19	13	22	17.6
146	Main condensate, °F	75	80	75	80	81	90	77	95.5	78	93.5	80	98.5
146a	Mn.turb.exh.minus mn.condensate, °F	6	8	17	13	19	9	34	13	27	12	36	14
147	Mn.condensate minus mn.ovbd.disch., °F	10	15	2	16	3	19	(-)8	20.5	0	22.5	(-)2	22.9
148	Dynamo condenser injection, °F	61	56	62	55	62	56	62	59.5	60	59	60	59
149	Dyn.cond.overboard discharge, °F	73	67	76	70	67	73	77	66.5	62	64	65	64
150	Dyn.cond.condensate, °F	95	106	88	105	83	109	95	95	80	90	80	91
151	Dyn.cond.circ.water rise °F	11	9	14	15	5	18	16	7	2	6	3	5
152	Dyn.condensate minus dyn.cond. overboard discharge, °F	21	38	12	38	16	36	18	25.5	18	31	17	27
153	Circ.water to mn.gen.air coolers, °F	60	58	61	54	62	58	62	59	59	56	60	57.5
154	Circ.water from mn.gen.air cool., °F	100	81	97	82	90	85	92	85	85	84	74	80

162	Dyn.cond.circ.water rise	°F	11	9	14	15	5	15	15	7	2	5	5
163	Dyn.condensate minus dyn.cond. overboard discharge,	°F	21	36	12	36	14	36	18	26.5	18	31	17	27
163	Circ.water to mn.gen.air coolers,	°F	60	58	61	54	62	58	62	59	59	56	60	57.5
164	Circ.water from mn.gen.air cool.,	°F	100	81	97	82	98	85	92	85	89	84	74	85
165	Circ.water rise,mn.gen.air cool.,	°F	40	23	36	28	36	25	30	26	30	28	14	28
166	Cir.water to mn.motor air coolr.,	°F	--	63	56	60	57	64	56	59	54	56	54	58
167	Cir.water from mn.str.air coolr.,	°F	--	83	80	82	70	90	62	76	66	70	62	70
168	Cir.water rise,mn.str.air coolr.,	°F	--	20	24	22	13	26	6	17	14	12	8	12
169	Cir.water to main turbo-gen.lubr. oil coolers,	°F	80	57.5	61	55	62	58	61	59	59	56	60	58
160	Cir.water from main turbo-gen. lubricating oil coolers,	°F	74	56	64	58	68	62	63	72	62	68	62	67
161	Cir.water rise main turbo-gen. lubricating oil coolers,	°F	14	.5	3	3	3	4	2	13	3	7	2	9
168	Cir.water to main motor & thrust block lubr.oil coolers,	°F	--	--	56	60	57	64	56	59	54	59	54	58
169	Cir.water from main motor &thrust block lubr.oil coolers,	°F	--	--	--	--	67	--	68	64	62	74	61	68
164	Cir.water rise,main motor&thrust block lubr.oil coolers,	°F	--	--	--	--	10	--	9	27	8	16	7	10
168	Lub.oil to mn.tur-gen.oil coolr.,	°F	114	109	115	151	115	151	110	122	110	126	118	122
166	Lub.oil from mn.tur-gen.oil clr.,	°F	108	82	86	90	88	93	90	84	86	101	87	107
167	Drop in lub.oil,mn.T-gen.oil clr.,	°F	6	27	27	41	27	38	20	38	24	25	21	25
168	Lub.oil to mn.str.Athrust oil clr.,	°F	96	96	95	101	89	99	93	105	98	102	106	105
169	Lub.oil from mn.str.Athr.oil clr.,	°F	98	96	85	96	80	.94	81	98	80	98	88	89
170	Drop in lub.oil, main motor & thrust oil cooler,	°F	0	0	10	7	9	6	12	7	16	10	26	16
171	Lub.oil to 750 kw.set oil coolr.,	°F	121	123	120	126	120	127	120	126	120	124	126	124
172	Lub.oil from 750 kw.set oil clr.,	°F	110	114	110	116	120	116	118	117	116	114	118	114
173	Drop in lub.oil, 750 kw. set oil cooler,	°F	11	9	10	10	9	15	11	9	15	10	10	10
174	Warmest mn.turbo-gen.bearing,	°F	132	128	132	149	136	158	156	160	150	150	144	150
175	Warmest mn.motor bearing,	°F	102	100	108	118	100	114	105	113	108	116	116	122
176	Warmest propeller thrust bearg.,	°F	104	104	104	129	110	118	107	110	116	114	120	118
177	Warmest 750 kw.turbo-gen.bearg.,	°F	137	130	130	150	139	130	139	150	157	150	139	150

*Estimated from temperatures of feed tanks.

TABLE I (CONTINUED)

NOV. - DEC. 1938

TABULATED TRIAL DATA

FINAL MACHINERY TRIALS

U.S.S. LEXINGTON & SARATOGA

ITEM NO.	ITEM	4 HRS AT 4000 SHP LEX	4 HRS AT 4000 SHP SARA	4 HRS AT 144 RPM LEX	4 HRS AT 144 RPM SARA	4 HRS AT 168 RPM LEX	4 HRS AT 168 RPM SARA	4 HRS AT 208 RPM LEX	4 HRS AT 208 RPM SARA	4 HRS AT 235 RPM LEX	4 HRS AT 235 RPM SARA	4 HRS AT 160000 SHP LEX	4 HRS AT 160000 SHP SARA
	TEMPERATURES (Continued):												
178	Main generator air supply, °F	104	--	95	--	95	--	98	--	94	--	89	--
179	Mn.gen.air disch.to coolers, °F	118	--	120	--	118	--	116	--	118	--	116	--
180	Main generator air rise, °F	18	--	25	--	23	--	18	--	21	--	27	--
181	Main motor air supply, °F	90	90	84	92	81	88	79	78	79	72	79	78
182	Mn.motor air disch.to coolers, °F	101	101	92	109	108	109	117	123	111	108	124	120
183	Main motor air rise, °F	11	8	6	17	27	21	38	47	32	36	45	46
184	Mn.gen.stator coils, maximum, °F	152	150	156	148	157	174	176	166	164	160	179	200
185	Mn.gen.stator coils, average, °F	126	120	144	140	159	159	148	142	147	154	148	181
186	Mn.gen.field coils, maximum, °F	186	190	218	200	220	220	226	230	220	220	220	248
187	Mn.gen.field coils, average, °F	117	190	203	200	217	218	224	223	229	192	228	212
188	Mn.motor stator coils, max., °F	123	120	134	146	152	160	182	209	140	164	174	220
189	Mn.motor stator coils, avg., °F	122	120	128	131	136	138	152	171	150	134	156	156
190	Fireroom,6 ft.above floor plate °F	94	79	96	81	94	83	89	79	89	79	88	77
191	Eng.rm.,6 ft.above floor plate °F	92	86	94	85	94	87	96	88	100	92	102	88
192	Motor rm.6'above floor plate °F	90	96	86	92	81	88	79	78	79	72	79	78
	REV.PER MIN. & DOUBLE STROKES:												
201	Main turbo-generator, average	1502	972	1502	1543	1778	1746	1222	1257	1515	1488	1725	1660
202	Main turbo-generator "C"	--	--	--	--	--	--	1197	1248	1492	1497	1756	1676
	"F"	1502	--	1502	--	1773	--	1247	--	1480	1482	1670	1648
	"E"	--	--	--	--	--	--	--	--	1540	1478	1744	1648
	"A"	--	972	--	1543	--	1746	--	1256	1547	1494	1729	1672
203	Main motor, average	90.23	89.20	144.00	144.00	161.96	161.88	226.46	226.06	272.89	275.07	312.13	280.86
204	Main motor - No. 1 shaft	90.26	89.21	144.14	144.02	162.19	161.88	226.74	226.09	270.86	275.88	310.68	311.69
	- No. 2 shaft	90.18	89.20	143.89	144.02	161.84	161.87	226.75	226.01	270.96	272.75	306.90	280.08
	- No. 3 shaft	90.19	89.27	143.96	143.97	161.84	161.89	226.21	226.01	270.60	272.38	307.98	280.29
	- No. 4 shaft	90.34	89.33	144.01	144.09	161.94	161.94	226.13	226.22	270.22	274.09	310.20	313.89
205	Mn.motor vent fan synchronous speed, average	486	461	792	792	891	889	686	686	792	791	888	889
206	[illegible]	5000	5000	5000	5000	5000	5000	5000	5000	5000	5000	5000	5000

	- No. 2 shaft	90.19	89.89	145.89	144.02	161.86	161.07	286.76	286.01	270.96	272.76	306.90	306.08
	- No. 3 shaft	90.19	89.87	145.98	143.97	161.84	161.09	286.81	285.91	270.60	272.38	307.90	305.99
	- No. 4 shaft	90.24	89.92	144.01	144.09	161.96	161.96	286.18	286.28	276.28	274.09	310.80	310.80
207	No.motor room fan synchronous speed, average	496	491	792	792	891	890	688	688	788	791	886	890
208	750 kw. turbines	5000	5000	5000	5000	5000	5000	5000	5000	5000	5000	5000	5000
209	Fuel oil booster pumps,avg.(30)	8	4½	14	6	11	8½	18	8	14	17	14	22
210	Fuel oil service pumps,avg.	172	178	266	290	348	418	896	727	600	782	786	616
211	Forced draft blowers, average	1008	848	984	1149	982	1437	1186	1821	1178	1150	1880	1488
212	Main feed pumps,average	2988	2180	2963	2075	2748	2060	2698	2227	2864	2714	2928	2266
213	Make up feed pumps, average(28)	Intermittent service at about 20 double strokes per minute											
214	Steam fire & bilge pumps - -on firemain,average(26)	--	18	--	16	--	20	--	17	--	18	--	17
	-on bilges, average(30)	16	--	27	29	26	30	19	27	22	26	14	25
215	Electric F & B pump on firemain	1199	--	1197	--	1297	--	1845	--	1846	--	1841	--
216	Main condenser circulating pumps	1148	426	1801	470	1198	717	1174	928	1287	886	1064	984
217	Main cond.,condensate pumps,avg.	1480	1600	1600	1410	1897	1460	1469	1290	1498	1390	1491	1378
218	Dyn.cond.,circulating pumps,avg.	1490	1690	1488	1862	1460	1390	1467	1487	1490	1848	1470	1622
219	Dyn.cond.condensate pumps, avg.	2860	2617	2489	2800	2870	2860	2486	2380	2487	2408	2482	2617
220	No.turbo-gen.lub.oil pumps,avg.	1089	810	1078	900	1081	988	1097	823	1016	618	1080	870
221	No.turbo-generator air and oil cooler and hydraulic vacuum pump supply pump, average	1874	900	1480	1080	1490	1000	1886	880	1189	980	1280	984
222	No.motor room lub.oil pumps,avg.	786	288	736	288	746	488	726	476	734	461	778	490
223	No. motor room air cool oil cooler pumps, average	1418	1480	1416	1546	1418	1842	1416	1608	1418	1881	1589	1884

TABLE 7
SHEET 5

*From main control board main generator field temperature indicators.

**From ships' electrical tachometers.

NOV. - DEC. 1928 TABLE 7 (CONTINUED)

FINAL MACHINERY TRIALS TABULATED TRIAL DATA U.S.S. LEXINGTON & SARATOGA

ITEM NO.	ITEM		4 HRS AT 4800 SHP LEX	4 HRS AT 4800 SHP SARA	4 HRS AT 144 RPM LEX	4 HRS AT 144 RPM SARA	4 HRS AT 162 RPM LEX	4 HRS AT 162 RPM SARA	4 HRS AT 226 RPM LEX	4 HRS AT 226 RPM SARA	4 HRS AT 273 RPM LEX	4 HRS AT 273 RPM SARA	4 HRS AT 180000 SHP LEX	4 HRS AT 180000 SHP SARA
	*MAIN PROPULSION ELECTRICAL:													
231	Mn.generator line volts, avg.		1989	2162	4023	4269	4750	5115	3870	3911	4095	4682	4950	5576
232	Mn.generator line volts,	"C"gen.	--	--	--	--	--	--	3880	4011	4022	4880	4972	5481
		"D"	1989	--	4023	--	4750	--	3889	--	4200	4638	4972	5586
		"B"	--	--	--	--	--	--	--	--	4172	4588	5000	5578
		"A"	--	2162	--	4269	--	5115	--	3811	3977	4555	4856	5611
233	Mn.generator line amperes, avg.		1782	2044	3223	3306	3812	3932	4508	4695	3846	3745	4755	4286
234	Mn.generator line amps.,	"C"gen.	--	--	--	--	--	--	4472	4414	3811	3786	4609	4233
		"D"gen.	1782	--	3223	--	3812	--	4535	--	3886	3830	4872	4322
		"B"gen.	--	--	--	--	--	--	--	--	3794	3733	4767	4300
		"A"gen.	--	2044	--	3306	--	3932	--	4977	3922	3622	4694	4236
235	Mn.gen.line power factor, avg.		.868	.490	.656	.610	.644	.610	.865	.868	.872	.855	.878	.860
236	Mn.gen.line power fact.,	"C"gen.	--	--	--	--	--	--	.866	.870	.870	.820	.878	.850
		"D"gen.	.868	--	.656	--	.644	--	.864	--	.870	.850	.878	.870
		"B"gen.	--	--	--	--	--	--	--	--	.870	.860	.879	.870
		"A"gen.	--	.490	--	.610	--	.610	--	.868	.860	.880	.878	.860
237	*Mn.gen.line indicated kw., total		3300	4190	13400	15880	20200	22000	52800	57945	94800	[illegible]	145800	142888
238	*Mn.gen.line ind.kw.,	"C"gen.	--	--	--	--	--	--	26400	28278	23400	24886	35900	35222
		"D"gen.	3300	--	13400	--	20200	--	26200	--	23300	25000	36400	36566
		"B"gen.	--	--	--	--	--	--	--	--	24000	25866	36400	35689
		"A"gen.	--	4190	--	15880	--	22000	--	29667	23800	23111	36600	34222
239	*Mn.gen.line kw.from √3xvolts x amps x P.F., total		3469	3751	14361	14912	20198	21250	52208	54935	95151	100886	145868	136439
240	*Mn.gen.line kw.from √3xvolts x amps x P.F.,	"C"gen.	--	--	--	--	--	--	25826	24679	23098	26131	36456	33829
		"D"gen.	3469	--	14361	--	20198	--	26388	--	24405	25179	36850	34755
		"B"gen.	--	--	--	--	--	--	--	--	23955	25216	36269	34048
		"A"gen.	--	3751	--	14912	--	21250	--	30254	23775	23520	34465	33829
241	*Main motor kw. input from shaft horsepower instruments, total		3751	3776	14756	15276	20780	21273	53449	55472	93799	97999	144165	142248
242	*Main motor kw. input from shaft horsepower instruments,	"C"gen.	--	--	--	--	--	--	26609	27356	23348	23898	35490	35150
		"D"gen.	3751	--	14756	--	20780	--	27141	--	23912	24604	36810	35444
		"B"gen.	--	--	--	--	--	--	--	--	23409	25006	36179	36228
		"A"gen.	--	3776	--	15276	--	21273	--	28117	23080	23694	35974	35236
243	*Calibration factor for mn.gen.line indic.wattmeters, avg.		1.80	.9012	1.1011	.9855	1.0287	.9670	1.0161	.9875	.9986	.9967	.9988	.9956
244	*Calibration factor for mn.gen.line indic.wattmeters,	"C"gen.	--	--	--	--	--	--	.9965	.9476	.9860	.9609	.9886	.9704
		"D"gen.	1.80	--	1.1011	--	1.0287	--	1.0258	--	1.0363	.9842	1.0050	.9751
		"B"gen.	--	--	--	--	--	--	--	--	.9837	1.0098	.9980	1.0094
		"A"gen.	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]

243	*Calibration factor for am.gen.line indic.-wattmeters , avg.	1.80	.9012	1.1011	.9836	1.0897	.9670	1.0161	.9878	.9936	.9967	.9938	.9955
244	*Calibration factor for am.gen.line indic.-wattmeters, "C"gen.	--	--	--	--	--	--	.9968	.9476	.9850	.9809	.9886	.9704
	"B"gen.	1.80	--	1.1011	--	1.0897	--	1.0088	--	1.0063	.9848	1.0080	.9781
	"D"gen.	--	--	--	--	--	--	--	--	.9857	1.0098	.9880	1.0064
	"A"gen.	--	.9012	--	.9836	--	.9670	--	.9478	1.0066	1.0089	.9886	1.0098
245	*Calibration factor for am.gen.line v3x volts x amps x PF kw., avg.	1.0815	1.0067	1.0847	1.0848	1.0886	1.0011	1.0896	1.0098	.9860	.9709	1.0065	1.0409
246	*Calibration factor for am.gen.line v3x volts x amps x PF kw., "C"gen.	--	--	--	--	--	--	1.0866	1.0886	.9675	.9053	1.0010	1.0488
	"B"gen.	1.0815	--	1.0847	--	1.0880	--	1.0888	--	.9898	1.0826	.9970	1.0896
	"D"gen.	--	--	--	--	--	--	--	--	.9898	1.0826	.9970	1.0896
	"A"gen.	--	1.0067	--	1.0845	--	1.0011	--	.9958	1.0068	1.0846	1.0878	1.0806
247	Main motor line amperes, average	488	888	782	885	980	987	2511	2988	1865	1788	2393	2819
248	Am.motor line amps.,No.17 motor	471	817	766	886	--	986	--	--	1806	1841	2888	2166
	No.11 motor	--	--	--	--	847	--	2006	2892	1788	1751	2288	2160
	No.27 motor	800	680	886	886	--	966	--	--	1866	1807	2660	2268
	No.21 motor	--	--	--	--	988	--	2611	2880	1866	1806	2666	2261
	No.37 motor	608	694	777	886	--	972	--	--	1888	1860	2600	2286
	No.31 motor	--	--	--	--	960	--	2688	2666	1861	1871	2606	2266
	No.47 motor	476	801	780	799	--	907	--	--	1988	1788	2606	2208
	No.41 motor	--	--	--	--	906	--	2194	2880	1889	1764	2666	2207
249	Main generator field volts.avg.	91	97	126	140	148	160	182	186	186	174	189	196
250	Am.generator field volts."C"gen.	--	--	--	--	--	--	176	188	184	178	188	196
	"B"gen.	91	--	126	--	148	--	188	--	188	168	166	198
	"D"gen.	--	--	--	--	--	--	--	--	181	168	164	888
	"A"gen.	--	97	--	140	--	160	--	188	178	186	186	199
251	Main generator field amps..avg.	864	442	884	602	602	692	710	787	674	678	658	771
252	Am.generator field amps.."C"gen.	--	--	--	--	--	--	686	787	884	718	613	788
	"B"gen.	864	--	884	--	602	--	737	--	871	678	680	761
	"D"gen.	--	--	--	--	--	--	--	--	898	608	689	788
	"A"gen.	--	442	--	602	--	692	--	786	860	706	661	780
253	Excitation indicator, average	8.8	9.8	6.4	8.7	4.8	8.8	4.8	2.8	6.8	7.8	2.8	8.1
254	Excitation indic.."C"gen.line	--	--	--	--	--	--	8.8	2.7	8.6	6.6	2.4	4.1
	"B"gen.line	8.8	--	6.4	--	4.8	--	4.2	--	6.4	8.7	1.8	6.8
	"D"gen.line	--	--	--	--	--	--	--	--	8.8	8.6	2.4	2.8
	"A"gen.line	--	9.8	--	8.7	--	8.8	--	2.8	6.8	9.1	8.8	7.4
255	*Booster motor-generator "motor" amps., avg.	187	196	194	172	188	184	89	84	181	121	188	86
256	*Booster motor-generator "motor" amps. for "C" generator	--	--	--	--	--	--	89	108	187	188	180	89
	"B" generator	187	--	194	--	188	--	89	--	188	171	187	87
	"D" generator	--	--	--	--	--	--	--	--	174	116	127	80
	"A" generator	--	196	--	172	--	184	--	64	174	180	189	71
257	Main motor efficiency(from tables)	.971	.966	.988	.986	.969	.980	.967	.987	.961	.969	.961	.968

*All observed data from switchboard instruments unless otherwise noted.
**Items 237 to 240 include power input to main motor vent fans; items 241 and 242 do not.
***Values given show "pump back" to d.c.sup.bus of "motor" and acting as a generator for bucking action of booster set on 240 volt d.c.supply.

TABLE 7
SHEET 4

NOV. - DEC. 1928 — TABLE V (CONTINUED) — U.S.S. LEXINGTON & SARATOGA

FINAL MACHINERY TRIALS — TABULATED TRIAL DATA

Item No.	Item	4 ENG AT 4800 SHP LEX	4 ENG AT 4800 SHP SARA	4 ENG AT 144 RPM LEX	4 ENG AT 144 RPM SARA	4 ENG AT 162 RPM LEX	4 ENG AT 162 RPM SARA	4 ENG AT 226 RPM LEX	4 ENG AT 226 RPM SARA	4 ENG AT 275 RPM LEX	4 ENG AT 275 RPM SARA	4 ENG AT 180000 SHP LEX	4 ENG AT 180000 SHP SARA
	SPEED, POWER & PROPELLER RPM:												
261	Average speed of vessel in knots corresponding to avg.prop.rpm.	10.88	10.48	16.91	16.90	19.10	18.90	26.48	26.66	31.66	31.67	34.24	33.91
	- 2 hr. water rate run	10.86	10.86	16.91	16.91	19.10	18.91	--	--	--	--	--	--
	- 4 hr. fuel consumption run	10.99	10.72	16.91	16.90	19.10	18.90	--	--	--	--	--	--
	- 4 hr. water rt.& fuel con.run	--	--	--	--	--	--	26.48	26.66	31.66	31.67	34.24	33.91
262	Tot.shaft horsepower,all shafts,avg.	4528	4586	18896	19068	26672	26514	68808	71067	120691	123738	180086	183456
	- 2 hr. water rate run	4442	4560	18785	19088	26131	25888	--	--	--	--	--	--
	- 4 hr. fuel consumpt.run	4668	4659	18066	18976	25742	26495	--	--	--	--	--	--
	- 4 hr. water rt.& fuel con.run	--	--	--	--	--	--	68808	71067	120691	123738	180086	183456
263	Avg.shaft hp.,No.1 shaft	1032	1184	4256	4824	5901	6638	18090	17329	28746	30275	45806	45588
	- 2 hr. water rate run	1081	1067	4415	4806	6176	6630	--	--	--	--	--	--
	- 4 hr. fuel consumpt.run	1032	1162	4179	4807	5762	6684	--	--	--	--	--	--
264	Avg.shaft hp.,No.2 shaft	1173	1130	4796	4800	6736	6625	18420	17735	30789	31098	46979	48008
	- 2 hr. water rate run	1178	1068	4896	4873	6906	6649	--	--	--	--	--	--
	- 4 hr. fuel consumpt.run	1172	1161	4746	4765	6651	6613	--	--	--	--	--	--
265	Avg.shaft hp.,No.3 shaft	1201	1165	4695	4980	6762	6946	18018	18901	30880	32181	46677	46791
	- 2 hr. water rate run	1163	1116	4610	4964	6684	6927	--	--	--	--	--	--
	- 4 hr. fuel consumpt.run	1220	1220	4634	4915	6802	6906	--	--	--	--	--	--
266	Avg.shaft hp.,No.4 shaft	1118	1086	4551	4814	6474	6510	16774	17122	30796	30688	44425	46474
	- 2 hr. water rate run	1075	1011	4648	4888	6566	6346	--	--	--	--	--	--
	- 4 hr. fuel consumpt.run	1140	1126	4606	4498	6626	6292	--	--	--	--	--	--
267	Avg.r.p.m. of propellers,all shf	90.25	89.30	144.00	144.05	161.96	161.90	226.46	226.06	275.25	275.07	312.13	309.84
	- 2 hr. water rate run	89.89	88.16	144.06	144.07	161.97	161.92	--	--	--	--	--	--
	- 4 hr. fuel consumpt.run	90.41	89.87	143.90	144.01	161.96	161.86	--	--	--	--	--	--
268	Avg.r.p.m. of propr.,No.1 shaft	90.36	89.31	144.14	144.08	162.19	161.88	226.74	226.08	276.36	273.38	310.88	311.60
	- 2 hr. water rate run	89.96	88.16	144.16	144.07	162.12	161.92	--	--	--	--	--	--
	- 4 hr. fuel consumpt.run	90.51	89.86	144.13	143.99	162.22	161.86	--	--	--	--	--	--
269	Avg.r.p.m. of prop.,No. 2 shaft	90.18	89.29	143.90	144.08	161.86	161.87	224.75	226.01	270.96	272.75	306.30	308.08
	- 2 hr. water rate run	89.82	88.15	143.96	144.04	161.85	161.89	--	--	--	--	--	--
	- 4 hr. fuel consumpt.run	90.36	89.86	143.87	144.00	161.87	161.86	--	--	--	--	--	--
270	Avg.r.p.m. of prop.,No. 3 shaft	90.19	89.27	143.98	143.97	161.84	161.80	226.21	226.91	270.60	272.12	307.90	308.57
	- 2 hr. water rate run	89.86	88.15	143.90	144.02	161.89	161.88	--	--	--	--	--	--
	- 4 hr. fuel consumpt.run	90.38	89.83	143.94	143.95	161.81	161.78	--	--	--	--	--	--
271	Avg.r.p.m. of prop.,No. 4 shaft	90.24	89.35	144.01	144.09	161.96	161.96	226.12	226.23	275.25	274.09	315.00	315.69
	- 2 hr. water rate run	89.92	88.20	144.06	144.12	162.02	162.00	--	--	--	--	--	--
	- 4 hr. fuel consumpt.run	90.40	89.89	143.99	144.06	161.98	161.94	--	--	--	--	--	--
	AUXILIARY POWER:												
281	Avg. total d.c. power, kw.	813	967	905	865	908	1025	1180	1371	1250	1620	1425	1675
	- 2 hr. water rate run. kw.	740	950	875	950	875	919	--	--	--	--	--	--
	- 4 hr. fuel consumpt.run kw.	880	975	920	880	925	1075	--	--	--	--	--	--

AUXILIARY POWER:

281	Avg. total d.c. power, kw.	615	947	906	823	908	1085	1180	1871	1250	1425	1435	1470
	- 2 hr. water rate run. kw.	740	980	878	980	878	819	--	--	--	--	--	--
	- 4 hr. fuel consumpt.run kw.	880	975	950	860	925	1070	--	--	--	--	--	--
282	Avg.d.c.power for excitation, including booster set losses,kw.	39	50	78	105	96	137	298	565	360	559	476	697
	- 2 hr. water rate run, kw.	40	71	88	109	97	154	--	--	--	--	--	--
	- 4 hr. fuel consumpt.run, kw.	38	52	76	100	98	138	--	--	--	--	--	--
283	Avg.d.c.power absorbed by main gen. fields, not excitation, kw.	29	45	67	84	87	117	265	265	211	459	421	606
	- 2 hr. water rate run, kw.	30	56	73	91	86	115	--	--	--	--	--	--
	- 4 hr. fuel consumpt.run, kw.	28	57	64	80	87	118	--	--	--	--	--	--
284	Avg.d.c.power for engine auxil. kw.	50	78	85	96	81	95	98	204	80	186	177	219
	- 2 hr. water rate run, kw.	55	77	80	95	78	94	--	--	--	--	--	--
	- 4 hr. fuel consumpt.run, kw.	48	76	85	96	85	95	--	--	--	--	--	--
285	Avg.d.c.power for ship's service power, kw.	575	644	898	543	568	623	638	560	696	786	600	639
	- 2 hr. water rate run, kw.	627	652	872	573	500	544	--	--	--	--	--	--
	- 4 hr. fuel consumpt.run, kw.	699	649	858	529	597	665	--	--	--	--	--	--
286	Avg.d.c.power for ship's lighting, nat.(lights,not-gen.output), kw.	150	187	188	141	167	170	142	194	172	156	172	169
	- 2 hr. water rate run, kw.	150	150	240	175	200	145	--	--	--	--	--	--
	- 4 hr. fuel consumpt.run, kw.	150	176	160	125	150	188	--	--	--	--	--	--

STEAM CONSUMPTION:

291	Live steam consumption of test sm.turb.,water rt.run,lbs/hr.	67060	67815	202714	200488	276457	287620	388519	401102	516858	500884	488888	480786
292	Water rate of test sm.turb.,water rate run,lbs.of live steam per shaft horsepower hour	15.096	15.926	10.870	10.881	10.880	10.886	11.294	11.185	10.298	9.969	10.280	10.188
293	Water rate of test sm.turb.under contract r.p.m.& power requirements for water rate run, lbs. of live steam per s.hp.hr.	15.08	15.926	10.870	10.880	10.880	10.880	11.280	11.140	10.298	9,970	10.286	10.180
294	Contract guaranteed water rate corrected for steam,vac.,& turb. speed conditions existing during W.R.run,lbs.live steam per shaft horsepower hour	15.482	15.972	15.079	12.824	11.097	11.674	12.548	11.978	11.886	10.988	11.984	11.880
295	Ratio of actual to guaranteed water rate for W.R.run	.980	.998	.982	.946	.980	.989	.90	.988	.9978	.911	.9856	.978
296	Total steam consump.,live plus aux.exhaust steam, of test sm. turb.for 4 hr.single unit fuel consump.runs with aux.exhaust admission, lbs. per hour	88848	91194	209216	200884	208888	——	—	—	—	—	—	—

NOV. - DEC. 1928

FINAL MACHINERY TRIALS

TABLE 9 (CONTINUED)

TABULATED TRIAL DATA

U. S. S. LEXINGTON & SARATOGA

Item No.	Item	4 HRS AT 4500 SHP LEX	4 HRS AT 4500 SHP SARA	4 HRS AT 144 RPM LEX	4 HRS AT 144 RPM SARA	4 HRS AT 162 RPM LEX	4 HRS AT 162 RPM SARA	4 HRS AT 226 RPM LEX	4 HRS AT 226 RPM SARA	4 HRS AT 272 RPM LEX	4 HRS AT 272 RPM SARA	4 HRS AT 180000 SHP LEX	4 HRS AT 180000 SHP SARA
	FRESH WATER:												
301	*Make up feed, gals.per hour	711	806	715	822	647	649	1471	908	1288	1830	2130	2470
302	*Make up feed, gals.per naut.mile	86.1	81.4	42.8	80.9	34.9	35.8	48.8	22.2	40.0	37.7	62.8	78.9
303	Distilled by evaps., gals.per hr.	1000	1967	1070	1617	1066	2700	1000	2250	1075	1650	Secured	2823
	- distilled to	Res.feed	Ship's tk.	R.F.	Res. feed	R.F.	R.F.	R.F.	R.F.	R.F.	S.T.	--	R.F.
	FUEL CONSUMPTION:												
311	Gals.@ 60°F of fuel oil per hour as fired, 2 hr.water rate run	1269	1886	2720	2195	2888	2994	--	--	--	--	--	--
	4 hr.fuel cons. run	1280	1448	2471	2026	2084	2086	--	--	--	--	--	--
	4 hr.WR & FC run	--	--	--	--	--	--	6657	10565	13067	15289	22250	22774
312	Lbs.of fuel oil per hr., as fired 2 hr.water rate run	10086	12840	21706	22460	29069	32088	--	--	--	--	--	--
	4 hr.fuel cons. run	9708	11804	19721	22006	28908	31290	--	--	--	--	--	--
	4 hr. WR & FC run	--	--	--	--	--	--	68190	88298	111066	166076	179726	181282
313	Lbs.of fuel oil per hr.per cu.ft. of furnace volume, as fired 2 hr.water rate run	3.430	3.861	5.126	4.614	5.204	4.688	--	--	--	--	--	--
	4 hr.fuel cons. run	4.076	3.188	4.686	4.230	4.892	4.724	--	--	--	--	--	--
	4 hr. WR & FC run	--	--	--	--	--	--	6.429	6.776	6.887	7.464	10.608	10.268
314	Lbs.of fuel oil per hr.per sq.ft. of water htg.surf., as fired 2 hr.water rate run	.3528	.2748	.4289	.4628	.4204	.4688	--	--	--	--	--	--
	4 hr.fuel cons.run	.3626	.3099	.4062	.4160	.4406	.4686	--	--	--	--	--	--
	4 hr. WR & FC run	--	--	--	--	--	--	.6061	.6680	.6178	.7826	.9866	1.0071
315	Lbs.of fuel oil per shaft horsepower hour, as fired 2 hr.water rate run	2.468	2.908	1.167	1.222	1.073	1.160	--	--	--	--	--	--
	4 hr.fuel cons.run	2.152	2.409	1.092	1.222	1.006	1.161	--	--	--	--	--	--
	4 hr. WR & FC run	--	--	--	--	--	--	.7911	1.158	.9006	1.008	.9669	.9871
316	Lbs.of fuel oil per nautical mile, as fired 2 hr. water rate run	1066	1171	1284	1806	1469	1441	--	--	--	--	--	--
	4 hr.fuel cons.run	917	1078	1106	1286	1586	1486	--	--	--	--	--	--
	4 hr. WR & FC run	--	--	--	--	--	--	2861	2026	2869	2461	2869	2846
317	Lbs.of fuel oil per hr.for fuel cons.run on basis of 18500 Btu.per lb.oil	9978	11871	19006	22122	28000	30126	48826	77108	107227	121867	172006	176411
318	Lbs.of fuel oil per hr.per cu. ft.furnace vol.for fuel cons. run on basis 18500 Btu.oil	2.988	3.066	4.407	4.080	4.788	4.846	6.217	6.514	6.288	7.126	10.277	9.896

319 Lbs.of fuel oil per hr.per sq. ft.water htg.surf.for fuel

	same.run on basis of 19800 Btu.per lb.oil	9978	11981	19966	23952	29939	30156	44929	59186	107887	143849	179811	179811
218	Lbs.of fuel oil per hr.per sq. ft.surface tot.for fuel cons. run on basis 19800 Btu.oil	2.998	3.066	4.687	6.050	6.762	6.958	6.519	8.536	6.529	7.185	10.227	9.096
219	Lbs.of fuel oil per hr.per sq. ft.water htg.surf.for fuel cons.run,basis 19800 Btu.oil	.9708	.2897	.4886	.6006	.4468	.4468	.5988	.6998	.8987	.9969	.9687	.9969
220	Lbs.of fuel oil per shaft hp.hr. for fuel cons.run,19800 Btu.oil	2.004	2.2806	1.088	1.187	.9714	1.187	.9968	1.1198	.8984	.9488	.9882	.9197
221	Lbs.of fuel oil per nautical mile for fuel cons.run,19800 Btu.oil	886.8	1084.6	1134	1088.7	1309	1804	3478	2965	3987	3851	3841	3148
222	Lbs.of fuel oil per naut.mile under contract F.P.S.S power requirements for fuel cons. run on basis of 19800 Btu.oil	866.9	1094.6	1139	1099	1309	1610	3660	3060	3980	3980	--	--
223	Lbs.of fuel oil per naut.mile guaranteed in contract for fuel cons.run on basis of 19800 B.T.U. oil	No guar.	No.guar.	1811	1811	1466	1459	3098	3098	4130	4130	No guar.	No guar
224	Ratio of actual to guaranteed fuel per naut.mile under contract trial conditions	--	--	.764	.893	.777	.896	.818	.966	.918	.926	--	--
225	B.t.u.fired per hr.(mils.by 10^6) 2 hr. water rate run	388.1	288.9	407.9	476.0	587.7	883.6	--	--	--	--	--	--
	4 hr. fuel consumption run	288.8	216.8	310.7	489.8	487.6	587.4	--	--	--	--	--	--
	6 hr. water rate & fuel cons.run	--	--	--	--	--	--	1868.7	1846.6	8096.9	8266.1	8979.0	8681.0
226	B.t.u.fired per hour per sq.ft. furnace vol. 2 hr. water rate run	44878	88889	96806	84608	99646	97886	--	--	--	--	--	--
	4 hr. fuel cons.run	78897	88870	87610	79844	98096	88888	--	--	--	--	--	--
	6 hr. 88 & 90 run	--	--	--	--	--	--	161881	187016	128408	140977	188480	188884
227	B.t.u.fired per hour per sq.ft. water htg.surface 2 hr. water rate run	6977	8166	9968	8489	9885	9488	--	--	--	--	--	--
	4 hr. fuel cons.run	7880	8886	8886	7888	8660.	9704	--	--	--	--	--	--
	6 hr. 88 & 90 run	--	--	--	--	--	--	13610	18468	11616	18767	18718	18884
228	B.t.u.fired per shaft hp.hr. 2 hr. water rate run	44180	44976	21744	21888	20000	21947	--	--	--	--	--	--
	4 hr. fuel cons.run	40068	44417	20886	20148	19989	20178	--	--	--	--	--	--
	6 hr. 88 & 90 run	--	--	--	--	--	--	16688	21707	17886	18618	18178	16618
	[redacted]:												
229	Overall efficiency,fuel to shaft 2 hr. water rate run	.2888	.2667	.1171	.1086	.1861	.1161	--	--	--	--	--	--
	4 hr. fuel cons. run	.2888	.2849	.1861	.1101	.1868	.1148	--	--	--	--	--	--
	6 hr. 88 & 90 run	--	--	--	--	--	--	.1866	.1176	.2470	.1888	.1601	.1878

*From soundings of all reserve feed bottoms. Due to drain collecting water being piped to various bottoms all bottoms have to be sounded giving only approximate results for runs of only several hours duration.

[redacted]

TABLE 8

TABULATED TRIAL DATA

SPECIAL ACCEPTANCE TRIALS OF DECEMBER 1929-JANUARY 1930 FOR TEST OF MAIN PROPULSION MACHINERY WITH REDESIGNED FOUR BLADE MAIN PROPELLER MOUNTED - FOUR HOURS AT FULL POWER - U.S.S. LEXINGTON & SARATOGA

ITEM NO.	ITEM	LEXINGTON	SARATOGA
	GENERAL DATA:		
1	Speed, knots from Nov.-Dec.1928 standardization	35.66	32.90
4	Duration of run, hours	4	4
5	Time commencing run	0000	0800
6	Date	Jan.22,1930	Dec.20,1929
8	Locality	So. Calif. Coast	So. Calif. Coast
9	Days out of dock	Less than 1	Less than 1
7	Displacement	Service	Service
8	Trim by the stern, at anchor equivalent	1'-6"	0'-11"
9	Sea	Smooth	Smooth
10	Relative wind - velocity, knots - bearing	31 Ahead	25 Ahead
11	Barometer, inches Hg.	30.21	30.11
12	Outside air temperature, degrees F.	59	59
13	Temperature of sea, degrees F.	57.1	57.9
	MACHINERY SET-UP:		
21	Number of boilers	16	16
22	Number of main turbo-generators	4	4
23	Number of main motors	8	8
24	Main motor pole combination	32	32
25	Number of forced draft blowers Total number of hand nozzle valves open	32 112	32 --
26a	Burner register opening, fraction of total area per burner	.53	--
30	Number of burners	112	128
30a	Size of sprayer plates	3006	

	Total number of hand nozzle valves open	118	--
26a	Burner register opening, fraction of total area per burner	.55	--
30	Number of burners	115	129
30a	Size of sprayer plates	8000	
43	Number of booster motor-generators	4	4
56	Disposition of auxiliary exhaust steam -		
	Main turbines to which admitted	All	All
	Stage to which admitted	10th	10th
	Main feed water heaters to which admitted	All	All
	Distilling plant units to which admitted	All	All
69	Disposition of drains -		
	High pressure to	Feed heaters	Feed heaters
	Low pressure to	Feed tanks	Feed tanks
	Feed heater to	Feed tanks	Feed tanks
	1st effect evaporator to	Dynamo condenser	Dynamo condenser
	Fuel oil heater	Feed tanks	Feed tanks

DIMENSIONAL DATA:

61	Total connected boiler water heating surface, sq.ft.	180000	180000
62	Total connected boiler superheating surface, sq.ft.	18788	19484
63	Total connected boiler furnace volume, cubic feet	16944	17664
73	Total connected main turbine capacity, rated horsepower	160872	160872
76	Total connected main motor capacity, rated horsepower	180000	180000

FUEL OIL CHARACTERISTICS:

81	Specific gravity, degrees A.P.I., about	14.5	14.5
84	Pounds per gallon at 60 °F, about	7.86	7.86
85	Viscosity, 4 degrees Engler at, estimated	2000	2000
86	Water and sediment, per cent, estimated	0.5	0.5
88	Flash point, degrees F., estimated	225	225
89	Source	California	California
89	Heat content, B.t.u. per pound as fired, estimated	18800	18800

TABLE 4

TABLE 8 (CONTINUED)

TABULATED TRIAL DATA

SPECIAL ACCEPTANCE TRIALS OF NOVEMBER 1939-JANUARY 1940 FOR TEST OF MAIN PROPULSION MACHINERY WITH SUPERHEATED STEAM USING NEW BOILER DESIGN - SHIPS BUILT AT NEW YORK - U.S.S. LEXINGTON & SARATOGA

ITEM NO.	ITEM	LEXINGTON	SARATOGA
	PRESSURE:		
91	Steam at boiler, pounds per sq. in., gage	861	868
98a	Steam at main turbine throttles(just ahead of main turbine steam strainers for Lexington and at main line at main turbine take-offs for Saratoga), pounds per sq. in., gage	841	876
96	Steam chest, main turbine, pounds per sq.in.,gage	838	868
98	1st stage, main turbine, pounds per sq. in., gage	182	184
96	9th stage, main turbine,aux.exhaust admission belt, lbs. per sq. in.,gage, or inches Hg.,vacuum	0.8#	7.8#
97	10th stage, main turbine, aux. exhaust admission belt, lbs. per sq. in.,gage, or inches Hg.,vacuum	1.7#	7.5#
98	Main turbine exhaust, inches Hg.,vacuum(gage board)	26.8"	28.0"
101	Auxiliary exhaust steam at main turbine,lbs.per sq.in.,Gg.	11.7	11.7
104	Firerooom air, inches water	6.4	6.3
106	Fuel oil at burner manifold, pounds per sq. in., gage	868	844
106a	Steam to fuel oil service heaters, lbs. per sq. in.,gage	168	35
109	Main turbine lub. oil pump disch.,lbs per sq.in.,gage	--	--
110	Main motor room lub. oil to bearings,lbs.per sq.in.,Gg.	8.5	8
111	Lubricating oil to main turbine governor, lbs. gage	86	76
112	Lub. oil to main turbo-gen. bearing supply line, lbs.Gg.	27	26
113	Lub. oil at main turbo-gen. bearings, lbs.per sq. in.,Gg.	14	13
	TEMPERATURE:		
131	Steam from boiler superheater, degrees F.	467	466
136	Superheat at boiler, degrees F.	81	78

	TEMPERATURES:		
131	Steam from boiler superheater, degrees F.	469	488
132	Superheat at boiler, degrees F.	51	73
135	Main turbine steam chest, degrees F.	440	490
136	Superheat at main turbine steam chest, degrees F.	55	80
137	Main turbine exhaust, degrees F.	111	109.5
139	Fuel oil booster pump suction, degrees F.	81	86
140	Fuel oil to burners, degrees F.	201	185
142	Main feed water heater outlet, degrees F.	208	221
143	Main injection, degrees F.	59.1	57.9
144	Main condenser overboard discharge, degrees F.	76.5	74.0
154	Circulating water from main generator air cooler, deg.F.	63	97
157	Circulating water from main motor air cooler, degrees F.	61	78
160	Circ.water from main turbo-gen. lub.oil cooler, deg.F.	59	99
163	Circulating water from main motor and thrust block lubricating oil cooler, degrees F.	60	79
165	Lubricating oil to main turbo-gen. oil cooler, deg. F.	115	137
166	Lubricating oil from main turbo-gen.oil cooler, deg. F.	90	101
168	Lub.oil to* main motor and thrust oil cooler, degrees F.	104	106
169	Lub.oil from main motor and thrust oil cooler, deg. F.	82	98
174	Warmest main turbo-generator bearing, degrees F.	156	164
175	Warmest main motor bearing, degrees F.	115	122
176	Warmest propeller thrust bearing, degrees F.	111	104
178	Main generator air supply(from coolers), degrees F.	85	--
179	Main generator air discharge to coolers, degrees F.	110	96
181	Main motor air supply, degrees F.	72	76
182	Main motor air discharge to coolers, degrees F.	119	98
184a	Main generator stator coils highest temperature indicator coil average for trial, degrees F.	(A-1)178	(B-1)210
185	Main generator stator coils, average, degrees F.	154	186
186a	Main generator field coils highest temperature indicator average for trial, degrees F.	(A)250	(A)244

*See item 84

TABLE 8 (CONTINUED)

TABULATED TRIAL DATA

SPECIAL ACCEPTANCE TRIALS OF DECEMBER 1929-JANUARY 1930 FOR TEST OF MAIN PROPULSION MACHINERY WITH REDESIGNED FOUR STAGE MAIN TURBINE HEADERS - FULL POWER AT SEA TRIAL - U.S.S. SARATOGA & SARATOGA

ITEM NO.	ITEM	LEXINGTON	SARATOGA
	TEMPERATURES (CONTINUED):		
187	Main generator field coils by temp. indic., average, °F.	220	216
188a	Main motor stator coils highest temperature indicator coil average for trial, degrees F.	(44-1)156	(44-8)171
189	Main motor stator coils, average, degrees F.	147	151
190	Firerooms, 6 feet above floor plates, degrees F.	87	80
191	Engnerooms, 6 feet above floor plates, degrees F.	98	98
192	Motor Rooms, 6 feet above floor plates, degrees F.	75	75
	REVOLUTIONS PER MINUTE:		
201	Main turbo-generator, average, (from electrical tachom.)	1668	1589
202	Main turbo-generator "C"	1735	1600
	"D"	1648	1598
	"B"	1647	1586
	"A"	1706	1543
203	Main motor, all shafts average	301.6	287.3
204	Main motor - No. 1 shaft	298.5	288.9
	- No. 2 shaft	295.8	282.8
	- No. 3 shaft	294.6	286.5
	- No. 4 shaft	307.5	292.7
211	Forced draft blowers, average	1376	1396
	MAIN PROPULSION ELECTRICAL:		
231	Main generator line volts, average	4875	4645
232	Main generator line volts, "C" generator	4888	4676
	"D" generator	4872	4572
	"B" generator	4756	4672
	"A" generator	4942	4650
233	Main generator line amperes, average	4816	4316
234	Main generator line amperes, "C" generator	4478	4041
	"D" generator	4897	4272
	"B" generator	4830	4211

		"B" generator	4796	4680
		"A" generator	4942	
233	Main generator line amperes, average		4516	4514
234	Main generator line amperes,	"C" generator	4478	4061
		"D" generator	4537	4672
		"B" generator	4530	4511
		"A" generator	4511	4432
236	Main generator line power factor, average		97.7	97.4
236	Main generator line power factor,	"C" generator	97.6	96.0
		"D" generator	97.8	90.0
		"B" generator	98.3	97.2
		"A" generator	97.8	96.8
237	Main generator line indicated Kilowatts, total		138600	134000
238	Main generator line ind. kw.,	"C" generator	34300	31800
		"D" generator	34300	28800
		"B" generator	34300	31900
		"A" generator	35700	32300
247	Main motor line amperes, average		2270	2166
248	Main motor line amperes,	No. 1F motor	2261	2067
		No. 1A motor	2256	2061
		No. 2F motor	2304	2160
		No. 2A motor	2300	2175
		No. 3F motor	2278	2217
		No. 3A motor	2311	2211
		No. 4F motor	2285	2208
		No. 4A motor	2255	2159
249	Main generator field volts, average		156	166
250	Main generator field volts,	"C" generator	145	170
		"D" generator	156	152
		"B" generator	156	177
		"A" generator	164	163
251	Main generator field amperes, average		668	665
252	Main generator field amperes,	"C" generator	660	690
		"D" generator	680	661
		"B" generator	665	661
		"A" generator	670	641
253	Exciter indicator, average		5.6	4.6
254	Exciter indicator,	"C" generator line	5.5	4.5
		"D" generator line	5.0	4.5
		"B" generator line	5.0	5.0
		"A" generator line	6.8	5.6

"All observed data from switchboard instruments unless otherwise noted."

TABLE 8
SHEET 3

TABLE 8 (CONTINUED)

SPECIAL ACCEPTANCE TRIALS OF DECEMBER 1939-JANUARY 1940 FOR TEST OF MAIN PROPULSION MACHINERY WITH REDUCTION GEAR DRIVE AND TURBINE BLADING - CONE SHAFT AT NEW YORK - U.S.S. [illegible] & [illegible]

ITEM NO.	ITEM	LOCATION	REMARKS
	SPEED, POWER, & PROPELLER R.P.M.:		
261	Average speed of vessel in knots corresponding to average propeller r.p.m. from Nov.- Dec. 1938 standardizations	33.46	32.79
262	Total shaft horsepower, all shafts, accepted, average	176006	158050
262a	Total s.hp., all shafts, as calculated from corrected indicated wattmeter readings	176556	160096
262b	Total s.hp., all shafts, as calculated from corrected voltmeter, ammeter, & P.F. meter readings	175455	156963
263	Average s.hp., No. 1 shaft, accepted	45060	39868
263a	Average s.hp., No. 1 shaft, from wattmeters	45871	40861
263b	Average s.hp., No. 1 shaft, from volts, amps., P.F.	44029	38435
264	Average s.hp., No. 2 shaft, accepted	43886	37310
264a	Average s.hp., No. 2 shaft, from wattmeters	45871	39826
264b	Average s.hp., No. 2 shaft, from volts, amps., P.F.	43961	36881
265	Average s.hp., No. 3 shaft, accepted	43554	40748
265a	Average s.hp., No. 3 shaft, from wattmeters	45464	41297
265b	Average s.hp., No. 3 shaft, from volts, amps., P.F.	43804	40199
266	Average s.hp., No. 4 shaft, accepted	44796	41360
266a	Average s.hp., No. 4 shaft, from wattmeters	45381	41680
266b	Average s.hp., No. 4 shaft, from volts, amps., P.F.	44341	41061
267	Average r.p.m. of propellers, all shafts average	301.6	287.3
268	Average r.p.m. of propeller, No. 1 shaft	302.3	288.9
269	Average r.p.m. of propeller, No. 2 shaft	300.8	288.8
270	Average r.p.m. of propeller, No. 3 shaft	304.6	288.8
271	Average r.p.m. of propeller, No. 4 shaft	297.5	286.7
	FEEDWATER:		
301	Make up feed, gallons per hour ..	1869	2869
302	Make up feed, gallons per nautical mile	56.4	109.6

FEED WATER:

301	*Make up feed, gallons per hour ..	1998	3580
302	*Make up feed, gallons per nautical mile	26.4	103.6

FUEL CONSUMPTION:

310	Gallons per hour of fuel oil, as fired, from meter readings uncorrected for temperature and meter calibration	22449	10904
311	Gallons of fuel oil at 60°F per hour, as fired	22830	10697
312	Pounds of fuel oil per hour, as fired(sp.gr.estimated)	167200	168400
313	Pounds of fuel oil per hour per cu. ft. of furnace volumb, as fired(sp.gr.estimated)	10.5	8.60
314	Pounds of fuel oil per hour per sq. ft. of water heating surface, as fired(sp.gr.estimated)	.904	.871
315	Pounds of fuel oil per hour per shaft horsepower,as fired (Specific gravity of oil estimated)	1.61	.988
316	Pounds of fuel oil per nautical mile, as fired(sp.gr.est.)	2250	4029

Note: Trials conducted at maximum power attainable without exceeding name plate rating values of turbo-electric propulsion machinery.

*From soundings of all reserve feed bottoms. Results are only approximate for runs of such short duration.

The data for both vessels are included in a single table for a given series of trials and item numbers correspond in all tables for ease of reference.

The official trials of November-December, 1928, are most completely covered since they were the most important and most carefully conducted series of trials. Table 7, therefore, should be used as a standard.

Standardization data have not been included in tabular form since the speed-power-R.P.M. relation of the vessels is shown in detail in Figures 19 to 21 inclusive.

Table 4 shows operation with the originally designed turbine blading. The data were obtained by the ship's force of the *Lexington* with the ship's regular equipment of instruments, observations being taken on signal with the aid of the signalling system installed for the official trials.

Table 5 covers the May-June, 1928, official trials which were conducted with the 12th and 13th stage rotor blading of the main turbines removed (13th stage diaphragms left in place). This series of trials was conducted by the Board of Inspection and Survey with all trial gear in service.

Table 6 and Figure 8 cover the performance of the *Lexington* on her 30-knot endurance run from San Pedro, California, to Honolulu, Hawaii. This run was made soon after the May, 1928, official trials with the turbines operating without 12th and 13th stage rotor blading. The ship's regular equipment of instruments was used, the trial gear having been removed. Since the *Lexington* had been standardized in the course of her official trials only two weeks previous to this run, this standardization is used rather than the finally accepted standardization obtained in November, 1928.

Table 7 is for the official trials of November-December, 1928, with all trial gear in use. Just prior to these trials redesigned 12th and 13th stage blading had been installed in the main turbines of both vessels. This new blading, although of different section, was designed to the same water rate as the original blading.

Table 8 is for the official four-hour high-power endurance run conducted for each vessel after the installation of redesigned 10th

stage blading. In each case the ship's regular equipment of instruments was used, the trial gear having been removed previously.

The more important features of the trial procedure as called for in the contracts for the construction of the *Lexington* and *Saratoga* follow:

(a) Standardization:

Three runs at about each of the following speeds: 9, 12, 16, 18, 22, 25, 28, 30, 32.5 knots and five runs at highest speed attainable.

(b) 4500 S.H.P. 6-hour endurance run:

First two hours water rate; last four hours fuel consumption; one main turbo-generator; distilling plant in operation.

(c) 144 R.P.M. 6-hour endurance run:

First two hours water rate; last four hours fuel consumption; one main turbo-generator; distilling plant in operation.

(d) 162 R.P.M. 6-hour endurance run:

First two hours water rate; last four hours fuel consumption; one main turbo-generator; distilling plant in operation.

(e) 226 R.P.M. 4-hour endurance and fuel consumption run:

Water rate measured on test main turbine only; two main turbo-generators; distilling plant in operation.

(f) 273 R.P.M. 4-hour endurance and fuel consumption run:

Water rate measured on test main turbine only; four main turbo-generators; distilling plant in operation.

(g) 180,000 S.H.P. 4-hour endurance and fuel consumption run:

Water rate measured on test main turbine only; four main turbo-generators; distilling plant secured.

(h) Maximum power one-hour endurance run:

At discretion of Trial Board.

During the two-hour single turbo-generator water rate runs and the four-hour multiple turbo-generator fuel consumption runs no auxiliary exhaust was admitted to the test main turbine or its condenser. Auxiliary exhaust was admitted, however, to the test main turbine during the four-hour single turbo-generator fuel consumption runs and to all other than the test main turbine during the multiple turbo-generator fuel consumption runs.

Steam and fuel consumption guarantees are shown in the tables.

The water rate guarantees are based on the estimated propeller performance of Figure 7, on main turbine throttle steam at 265 pounds per square inch gage and 50 degrees F. superheat, and on a main turbine vacuum of 28.5 inches of mercury (30-inch barometer) for all runs up to and including the 273 R.P.M. run and a vacuum of 28.0 inches for the 180,000 S.H.P. run. Correction factors for variations in steam pressure, quality of steam, vacuum and propeller R.P.M. were allowed in the contracts as follows:

Steam pressure at turbine throttle:

For variations between 265 and 245 pounds gage, 1 per cent increase in guaranteed water rate for each 10 pounds reduction in steam pressure below 265 pounds gage.

Superheat in steam at turbine throttle:

1 per cent reduction in guaranteed water rate for each 13 degrees F. of superheat in excess of 50 degrees F. superheat. 1 per cent increase in guaranteed water rate for each reduction of 13 degrees F. of superheat below 50 degrees F. superheat.

Moisture in steam at turbine throttle:

2 per cent increase in guaranteed water rate for each 1 per cent of moisture.

Vacuum at turbine exhaust:

Run	Specified Vacuum (30-inch Barometer)	Increase in Guaranteed Water Rate per Inch Reduction in Vacuum Below Specified Vacuum
4500 S.H.P.	28.5	10 per cent
144 R.P.M.	28.5	8 per cent
162 R.P.M.	28.5	6 per cent
226 R.P.M.	28.5	6 per cent
273 R.P.M.	28.5	6 per cent
180,000 S.H.P.	28.0	6 per cent

Propeller R.P.M.:

1 per cent increase in guaranteed water rate for each 5 per cent reduction in propeller R.P.M. below 317 R.P.M. at 180,000 S.H.P. and 3½ per cent increase for each 5 per cent reduction below 91 R.P.M. at 4500 S.H.P.

In Tables 5 and 7 all water rates are for the test main turbines of each vessel. The contracts provided that during the multiple turbo-generator runs, the water rates of the other main turbines for the same operating conditions would be presumed to be the same as that of the test main turbine. To simplify the analysis of the turbine performance this extension has not been made in the foregoing tables. Comparison of the test main turbine performance with the overall guarantee values, however, has been included in Table 7.

Fuel consumption guarantees are based on oil having a heat content of 19,500 B.T.U. per pound.

For the May-June, 1928, official trials, in the case of the *Lexington*, the fuel consumption and endurance runs at 4500 S.H.P., 144 R.P.M., and 162 R.P.M. were reduced from four to two hours' duration. All other runs of both series of 1928 official trials were in accordance with the procedure called for in the contracts.

SUPPLEMENTARY PERFORMANCE DATA AND CURVES.

Since few Naval vessels have had their main propulsion and auxiliary machinery so thoroughly tested out during their fitting-out periods as these vessels, the more important data thus obtained have been summarized for inclusion in this compilation.

Due to the high power developed during the dock trials of the *Lexington* there are included Table 9 and Figure 9 showing propeller R.P.M. and shaft horsepower.

Because of the large capacity of the distilling plants and the unusually complete equipment of instruments fitted, the data obtained during the capacity tests are shown in Table 10. To more clearly show the feed sequence Figure 10 is included.

In the case of the *Lexington's* refrigerating plants special arrangements were made to measure the refrigerating capacity during the required operating tests. The data thus obtained are shown in Table 11.

For ready reference fuel consumption curves for each of the main turbo-generator-main motor combinations have been drawn based on the data obtained during the two series of 1928 official

TABLE 9

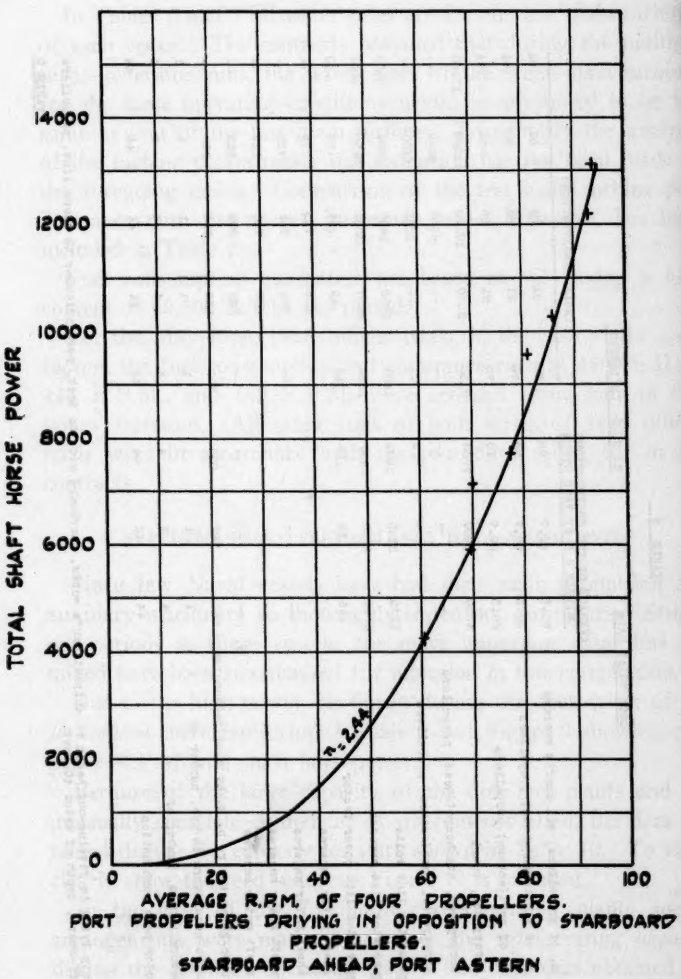
DOCK TRIAL DATA — U. S. S. LEXINGTON

	AVERAGE OF TWO OBSERVATIONS WITHIN 3 % OF EACH OTHER			SINGLE OBSERVATIONS				
R. p. m. of propellers, all shafts average	60.5	69.7	78	70	81	86	93	94
R. p. m. of starboard(ahead) propellers	60.5	69.7	78	70	81	86	93	94
R. p. m. of port(astern) propellers	60.5	69.7	78	70	81	86	93	94
Shaft horsepower, total	4230	5900	7700	7130	9570	10250	12200	13100
Shaft horsepower, starboard(ahead) propellers	--	--	--	--	--	4985	6000	--
Shaft horsepower, port(astern) propellers	--	--	--	--	--	5265	6200	--
Main generator line volts	1775	2125	2200	1900	2850	2400	2800	3000
Main generator line amperes	2362	2700	2900	2600	2900	3200	3450	3400
Main generator line power factor	.46	.56	.57	.56	.49	.64	.59	.60
No. 1 shaft motor amperes	--	--	--	--	725	725	775	--
No. 2 shaft motor amperes	--	--	--	--	725	700	725	--
No. 3 shaft motor amperes	--	--	--	--	600	700	725	--
No. 4 shaft motor amperes	--	--	--	--	725	800	825	--
Main turbo-generator r. p. m. by elec. tachometer	662	787	837	700	860	950	950	--
Main turbo-generators in use, number of	1	1	1	1	1	1	1	1
Main motors in use, number of	4	4	4	4	4	4	4	4
Main motor pole combination	44	44	44	44	44	44	44	44

Note:

Ship lying in fitting-out basin at shipbuilder's works, starboard side to dock, perpendicular to shore line, propellers off-shore, depth of water 40 feet.

TABLE 9



DOCK TRIAL PROPELLER PERFORMANCE
NOV. 21-23, 1927
U.S.S. LEXINGTON.

FIGURE 9.

TABLE 10

DISTILLING PLANTS - SUMMARY OF RESULTS - USS LEXINGTON - USS SARATOGA

Note:- All observations and calculations given below are average values.
All figures given in the test with modified feed sequence are
average values for both units.
For explanation of feed sequence see remarks at end of table.
Total evaporator heating surface per ship ---- 2880 sq.ft.
Type: Two modified Lillie quadruple effect units per ship.

Item No.	Item	LEXINGTON Original Feed Sequence Port	LEXINGTON Original Feed Sequence Stbd	LEXINGTON Modified Feed Sequence	SARATOGA Original Feed Sequence Port	SARATOGA Original Feed Sequence Stbd
1	Total capacity of plant (observed), gal/24 hr		97,000	------		105,000
2	Total capacity of plant (corrected), gal/24 hr		101,376	94,600		------
3	Salinity of distillate (chemical test), gr/gal		.42	.57		------
4	Salinity of brine overboard, 32nd		1.07	------		------
5	Press.-Asst.sub. to 1st. effect evaporators, lbs.g		7.0	------		12
6	Press.-1st effect tube nest, in vacuum	1.1	0.4	2.0/g	0	0
7	Temp.- 1st effect tube nest, F.	223.0	220.4	220.4	215.1	212.8
8	Press. 4th effect shell, in. vacuum	26.0	26.0	26.0	26.1	26.0
9	Temp.- 4th effect shell F	127.8	127.7	126.8	124.3	127.0
10	Total temp. drop F	96.3	100.7	126.4	90.8	85.8
11	Gal/24 hr/sq.ft.heat surf/F.total temp.drop	1.88	1.83	.918	2.32	1.80
12	Temp.-2nd. effect tube nest F	187.9	181.8	201.0	200.7	200.3
13	Temp.-3rd. effect tube nest F	169.6	173.3	177.3	188.0	190.7
14	Temp.-4th. effect tube nest F	140.4	148.3	157.2	170.3	167.8
15	Temp.-1st effect drains to heater F	207.7	206.0	218.0	210.3	208.1
16	Temp.-1st effect drains from heater F	160.6	159.0	168.0	167.8	169.0
17	Temp.-4th effect drains to heater F	148.0	148.4	155.4	168.8	164.8
18	Temp.-4th effect drains from heater, F	121.2	122.6	129.0	138.3	141.3
19	Temp.-Flash chambers, F	110.3	121.0	122.6	128.8	131.0
20	Temp.-Brine discharge overboard, F	187.7	129.5	121.8	130.7	131.4
21	Temp.-Dist.cond. circ. water disch., F	99.0	100.0	96.8	91.8	99.1
22	Temp.-Evap.feed disch. fr.dist. condsr., F	106.0	115.0	101.6	126.2	116.0
23	Temp.-Evap.feed disch. fr.4th.eff dr. htr., F	116.0	113.5	121.6	129.3	121.1
24	Temp.-Evap.feed disch. fr.1st.eff dr. htr., F	126.6	126.2	130.6	126.7	130.1
25	Temp.-Evap.feed disch. fr.air eject.condsr., F	120.0	122.4	130.4	141.8	149.7
26	Temp.-Evap.feed disch. fr.2nd.eff.vap.htr., F	142.0	144.0	156.8	161.8	168.8

Remarks: Original feed sequence: Feed to 4th effect, overflow to 3rd effect, overflow to 2nd effect, overflow to 1st effect, overflow to flash chamber, overflow to overboard.
Modified feed sequence: Feed to 4th effect, overflow to 3rd effect, overflow to 2nd effect, overflow to 1st effect, overflow to flash chamber, overflow to 4th effect, overflow to 3rd effect, overflow to 2nd effect, overflow to 1st effect, overflow to flash chamber, overflow to overboard.

25	Temp.-Evap.feed disch. fr.4th.eff dr. htr., °F	118.0		113.6		128.6		129.3		121.1	
26	Temp.-Evap.feed disch. fr.1st.eff dr. htr., °F	108.6		106.2		119.6		108.7		110.1	
27	Temp.-Evap.feed disch. fr.air eject.condsr., °F	108.0		105.4		119.4		101.8		107.7	
28	Temp.-Evap.feed disch. fr.3rd.eff.vap.htr., °F	148.0		144.0		154.0		151.5		150.9	
29	Temp.-Evap.feed disch. fr.2nd.eff.vap.htr., °F	161.4		170.6		173.0		179.5		165.5	
30	Temp.-Evap.feed disch. fr.1st.eff.vap.htr., °F	192.0		185.6		191.6		196.5		190.4	
31	Brine disch.overbd.,gal/hr.		4110			4340			4,370		
32	Press-Dist.circ. pump disch., lbs. #=		31.7			-----			32		
33	Press-Evap. feed pump disch., lbs. #=		39.7			-----			38		
34	Temp.-Dist.circ.pump injection, °F		68.6			86.6			72.7		
35	Temp.-Dist.circ.pump overbd., °F		68.1			86.8			81.5		
36	Temp.-Fresh water pump disch., °F		120.7			128.0			126.5		
37	H.P. input-brine circ. pumps	6.5		6.86		-----		6.66		6.16	
38	H.P. input-dist.circ. pump		13.4			-----			9.47		
39	H.P. input-evap. feed pump		2.89			-----			2.68		
40	H.P. input-brine disch. pump		5.85			-----			5.56		
41	H.P. input-dist. condst. pump		4.86			-----			3.77		
42	H.P. input-dist. f.w. pump		4.15			-----			3.69		

Remarks on Feed Sequence:

The original sequence of the evaporator feed circuit for each quadruple unit was as follows:-

1. Feed passes in distilling condenser.
2. Evaporator feed pump.
3. Fourth effect tube nest drain feed heater.
4. Air ejector condenser.
5. Third effect vapor feed heater.
6. Second effect vapor feed heater.
7. First effect vapor feed heater.
8. First effect tube nest drain feed heater.
9. First effect evaporator.

During the preliminary tests of the Lexington's distilling plant trouble was experienced in securing proper drainage of the 1st effect tube heats. From experimental runs conducted for the purpose of determining the cause of this trouble, it was found that the high temperature 1st effect drains were flashing in the drain line connected to the dynamo condensers, preventing proper drainage. The feed sequence was then modified as follows:-

1. Feed passes in distilling condenser.
2. Evaporator feed pump.
3. Fourth effect tube nest drain feed heater.
4. First effect tube nest drain feed heater.
5. Air ejector condenser.
6. Third effect vapor feed heater.
7. Second effect vapor feed heater.
8. First effect vapor feed heater.
9. First effect evaporator.

with the modified feed sequence no trouble was experienced in securing proper drainage of the first effect tube heats.

General Conditions during Tests:

Lexington test conducted with vessel at shipbuilder's dock, Boston Harbor. Evaporator tubes covered with moderate thickness of scale at time of test.

Saratoga test conducted with vessel at shipbuilder's dock, Delaware River(fresh water). Evaporator tubes free of scale at time of test.

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TABLE 11

REFRIGERATING PLANT TESTS - SUMMARY OF RESULTS - U.S.S. LEXINGTON

		FORWARD PLANT			AFTER PLANT	
	Number of compressors	2 (1 in service)			3 (2 in service)	
	Type	CO_2, vertical, single cylinder double acting motor driven.			CO_2, vertical, 2 cylinder, single acting motor driven.	
	Size	2½" bore, 8" stroke.			5½" bore, 5" stroke.	
	Rating of each plant in refrigeration per 24 hours.	6 tons (1 compressor in service)			12 tons (2 compressors in service)	
Item No.	Item	FORWARD PLANT			AFTER PLANT	
		24 hr. Test	4 hr. Test	8 hr. Test	24 hr. Reduced Capacity Test	8 hr. Full Capacity Test
1	Total refrigeration, tons/24 hr.	7.58	7.23	7.63	9.9	13.85
2	Refrigeration in Ice Tanks, tons/24 hrs.	----	----	----	1.76	3.6
3	Refrigeration, tons/24 hr., in scuttlebutts	.84	1.31	.546	1.24	2.85
4	Refrigeration in air cooler, tons/24 hr.	3.66	2.96	3.93	----	----
5	Refrigeration, tons/24 hr., in photo.lab. cooler (s.w.coil)	1.54	----	----	----	----
6	Refrigeration, tons/24 hr., in photo.lab. cooler (f.w.coil)	1.74	2.44	2.75	----	----
7	Remaining Refrigeration (Ref'd.Compts. and losses), tons/24 hrs.	----	----	----	6.9	7.65
8	Temperature of brine supply °F	21.8	25	23.2	−2.57	+2.3
9	Temperature of brine return, °F	29.7	27	28.9	+1.77	+6.3
10	Temperature rise of brine, °F	7.9	12	5.7	4.34	4.0
11	Brine circulation, total, G.P.M.	27.22	16.36	31.76	65.39	99.21
12	Specific gravity of brine	1.208	1.14	1.14	1.23	1.23
13	Weight of brine, lb./gal.	10.05	9.5	9.5	10.26	10.26
14	Specific heat of brine	.70	.775	.775	.68	.68
15	Temperature Air outside Refrigerated Compts., (average), °F	----	----	----	66.5	62.8
16	Temperature Meat Rooms, °F	----	----	----	9.9	10.4
17	Temperature Vegetable Rooms, °F	----	----	----	36.1	27.8
18	Temperature Butter and Egg Rooms, °F	----	----	----	30.5	30.0
19	Temperature Officers' Cooling Room, °F	----	----	----	33.4	25.0
20	Temperature Crew's Cooling Room, °F	----	----	----	34.1	36.0
21	Temperature of Fresh Water for ice making, F.	----	----	----	67.0	67.0
22	Temperature Ice Tank Brine, °F.	----	----	----	+1.15	+4.73
23	Total weight of Ice made, lbs.	----	----	----	2,754	468
24	Water cooled in photo.lab.cool.(s.w.coil) G.P.M.	2.16	----	----	----	----
25	Temp.water to photo lab.cool. (s.w.coil)°F	84.0	----	----	----	----
26	Temp.water from photo lab.cool (s.w. coil) °F	83.6	----	----	----	----
27	Temp. drop of water in cooler (s.w. coil),°F	80.4	----	----	----	----
28	Water cooled in photo.lb.cooler(f.w. coil) G.P.M.	2.44	4.31	4.42	----	----
29	Temp. water to photo.lab.cooler(f.w. coil), °F	86.6	88	86.4	----	----
30	Temp. water from photo lab.cooler (f.w. coil) °F	83.6	84	83.6	----	----
31	Temp. drop of water in cooler (f.w. coil), °F	3.0	4	2.8	----	----

27	Temp. drop of water in cooler (s.w. coil), °F	20.4	----	----	----	----
28	Water cooled in photo.lab.cooler(f.w. coil) G.P.M.	2.44	4.31	4.42	----	----
29	Temp. water to photo.lab.cooler(f.w. coil), °F	85.6	85	85.4	----	----
30	Temp. water from photo.lab.cooler (f.w.coil), °F	65.3	69.6	68.5	----	----
31	Temp.drop of water in cooler (f.w.coil), °F	20.3	15.4	16.9	----	----
32	Temp. brine in photo lab. cooler, °F	46.3	42.8	33.2	----	----
33	Brine circulation-air cooler, G.P.M.	19.61	6.54	21.39	----	----
34	Temp. brine return from air cooler, °F	57.4	54.4	38.8	----	----
35	Temp. rise of brine thru air cooler, °F	8.6	9.4	5	----	----
36	Water cooled in scuttlebutts, G.P.M.	4.68	.796	.548	.726	1.88
37	Temp. water to scuttlebutts, °F	89.5	89.6	90.2	90.3	90.4
38	Temp. water from scuttlebutts, °F	50.7	44.9	46.9	49.5	48.6
39	Temp. drop in scuttlebutts, °F	38.8	44.7	43.3	40.8	47.8
40	CO_2 suction press, lbs./sq.in	361	394	387	254.3	280
41	CO_2 discharge press, lbs. /sq.in	1158.9	1101	1090	1163.7	1187
42	Compressor R.P.M.	214	218	224	176.9	251.1
43	No. compressors in service	1	1	1	2	2
44	Compressor power input, total H.P.	20.4	21.7	21.8	34.35	30.15
45	Compressor H.P. per ton of refrigeration	2.7	3.0	2.78	3.47	3.42
46	Circ. water inlet temp., °F	84	82.1	82	84.4	84.6
47	Circ. water outlet temp. °F	88.7	88.7	88.4	85.64	87.0
48	Circ. water temp. rise, °F	4.7	6.6	6.4	1.24	2.4
49	Circulating Pump - Power input, H.P.	(From Sanitary System)			4.65	4.75
50	Brine circ. pump-power input, H.P.	2.04	1.91	2.15	5.8	6.4

- SPECIFICATIONS -

Forward Plant

The forward plant consists of two Brunswick-Kroeschell 6-ton, 24"x28" vertical, single cylinder, double acting, motor driven CO_2 compressors; one shell and coil type brine cooler (2 sections of 3 coils each); two CO_2 double pipe condensers (2 sections each); two brine circulating pumps and the necessary scale traps, separators, etc.

The circulating water supply for the CO_2 condensers is taken from the ship's sanitary system.

This plant serves the two forward scuttlebutts, a photographic laboratory fresh and salt water cooler, and an air cooler.

One CO_2 compressor, one brine circulating pump, one CO_2 condenser (2 sections) and one brine cooler section (3 coils) should produce concurrently for a period of 24 hrs. the following refrigeration, with circulating water at 85° F.

(a) - Photographic Laboratory Cooler

Cool 4 gal./min. salt water from 88° F. to 65° F. for one-half hour every alternate half hour.

Cool 1 gal./min.fresh water from 88° F. to 65° F. continuously.

(b) - Air Cooler-

Reduce the air temperature through the air cooler to, and maintain at, 65° F., when absorbing 21,000 B.T.U./hr.

(c) - Scuttlebutts -

Cool 600 gals/ 24 hrs. of drinking water (300 gal. per scuttlebutt) from 90° F. to 65°.

After Plant

The after refrigerating plant consists of three York 6-ton, 24"x28" vertical, twin cylinder, single acting, motor driven CO_2 compressors; one brine cooler (5 sections); one CO_2 condenser (3 sections); one ice making tank (3 sections); two brine circulating pumps; two CO_2 condenser circulating pumps; CO_2 dryer and the necessary traps, separators, etc.

This plant serves the refrigerated compartments on the 2nd, and 3rd decks, the ice making tanks and the three after scuttlebutts.

Any two CO_2 compressors with one CO_2 condenser, circulating pump, one brine circulating pump, two condenser sections and two brine cooler sections should produce concurrently for a period of 24 hrs., the following refrigeration with circulating water at 85 F., and outside air at 100 F.

(a) - Maintain the refrigerated compartments at the temperatures listed below.

Compartment	Total Volume	Temperature
Meat Rooms	2720 cu. ft.	15 F.
Vegetables	6884 cu. ft.	40 F.
Butter & Eggs	616 cu. ft.	40 F.
Crew's Cooling Rooms	1847 cu. ft.	40 F.
Officers' Cooling Rooms	3036 cu. ft.	40 F.

(b) - Cool 1000 gals. of drinking water per 24 hr. [illegible] gals. for each scuttlebutt

	Number of compressors		2 (1 in service)		3 (2 in service)	
	Type		CO_2, vertical, single cylinder double acting motor driven.		CO_2, vertical, 2 cylinder, single acting motor driven.	
	Size		8½" bore, 8" stroke.		9½" bore, 8" stroke.	
	Rating of each plant in refrigeration per 24 hours.		6 tons (1 compressor in service)		12 tons (2 compressors in service)	
Item No.	Item	FORWARD PLANT			AFTER PLANT	
		24 hr. Test	4 hr. Test	8 hr. Test	24 hr. Reduced Capacity Test	8 hr. Full Capacity Test
1	Total refrigeration, tons/24 hr.	7.58	7.23	7.83	9.9	13.63
2	Refrigeration in Ice Tanks, tons/24 hrs.	----	----	----	1.76	3.6
3	Refrigeration, tons/24 hr., in scuttlebutts	.64	1.31	.546	1.84	2.85
4	Refrigeration in air cooler, tons/24 hr.	3.66	2.96	3.93	----	----
5	Refrigeration, tons/24 hr., in photo.lab. cooler (s.w.coil)	1.54	----	----	----	----
6	Refrigeration, tons/24 hr., in photo.lab. cooler (f.w.coil)	1.74	2.44	2.75	----	----
7	Remaining Refrigeration (Ref'd.Compts. and losses), tons/24 hrs.	----	----	----	6.9	7.68
8	Temperature of brine supply °F	21.8	25	23.2	−2.57	+2.3
9	Temperature of brine return, °F	29.7	37	29.9	+1.77	+6.3
10	Temperature rise of brine, °F	7.9	12	6.7	4.34	4.0
11	Brine circulation, total, G.P.M.	27.22	16.36	31.76	65.39	99.21
12	Specific gravity of brine	1.205	1.14	1.14	1.25	1.23
13	Weight of brine, lb./gal.	10.05	9.5	9.5	10.36	10.25
14	Specific heat of brine	.70	.775	.775	.68	.68
15	Temperature air outside refrigerated Compts., (average), °F	----	----	----	66.5	62.6
16	Temperature Meat Rooms, °F	----	----	----	9.9	10.4
17	Temperature Vegetable Rooms, °F	----	----	----	36.1	37.8
18	Temperature Butter and Egg Room, °F	----	----	----	29.5	30.0
19	Temperature Officers' Cooling Room, °F	----	----	----	33.4	35.0
20	Temperature Crew's Cooling Room, °F	----	----	----	34.1	36.0
21	Temperature of Fresh Water for ice making, F.	----	----	----	67.0	67.0
22	Temperature Ice Tank Brine, °F.	----	----	----	+1.15	+4.75
23	Total weight of Ice made, lbs.	----	----	----	2,754	468
24	Water cooled in photo.lab.cool.(s.w.coil) G.P.M.	2.16	----	----	----	----
25	Temp.water to photo lab.cool. (s.w.coil)°F	64.0	----	----	----	----
26	Temp.water from photo lab.cool (s.w. coil) °F	65.6	----	----	----	----
27	Temp. drop of water in cooler (s.w. coil),°F	20.4	----	----	----	----
28	Water cooled in photo.lab.cooler(f.w. coil) G.P.M.	2.44	4.31	4.42	----	----
29	Temp. water to photo.lab.cooler(f.w. coil), °F	65.6	66	65.4	----	----
30	Temp. water from photo.lab.cooler (f.w.coil),°F	65.3	69.6	68.5	----	----
31	Temp.drop of water in cooler (f.w.coil),°F	20.3	15.4	16.9	----	----
32	Temp. brine in photo lab. cooler, °F	46.3	42.8	33.2	----	----
33	Brine circulation-air cooler, G.P.M.	19.61	8.54	21.37	----	----
34	Temp. brine return from air cooler, °F	27.4	34.4	28.2	----	----

33	Temp. water from photo.lab.cooler (f.w.coil), °F	80.5	15.4	16.9	----	----
34	Temp.drop of water in cooler (f.w.coil), °F	46.5	42.8	33.2	----	----
35	Temp. brine in photo lab. cooler, °F	19.61	8.54	21.37	----	----
36	Brine circulation-air cooler, G.P.M.	27.4	34.4	28.8	----	----
37	Temp. brine return from air cooler, °F	8.6	9.4	8	----	----
38	Temp. rise of brine thru air cooler, °F	4.68	.796	.342	.726	1.38
39	Water cooled in scuttlebutts, G.P.M.	89.5	89.6	90.2	90.5	90.4
40	Temp. water to scuttlebutts, °F	50.7	44.9	44.9	49.5	42.6
41	Temp. water from scuttlebutts, °F	38.0	44.7	45.3	40.8	47.8
42	Temp. drop in scuttlebutts, °F	361	384	397	254.5	220
43	CO_2 suction press, lbs./sq.in	1138.9	1101	1090	1143.7	1187
44	CO_2 discharge press, lbs. /sq.in	214	216	224	196.9	251.1
45	Compressor H.P.M.	1	1	1	2	2
46	No. compressors in service	20.4	21.7	21.8	34.38	30.15
47	Compressor power input, total H.P.	2.7	3.0	2.78	3.47	3.42
48	Compressor H.P. per ton of refrigeration	84	82.1	82	84.4	84.6
49	Circ. water inlet temp., °F	88.7	88.7	88.4	85.64	87.0
50	Circ. water outlet temp. °F	4.7	6.6	5.4	1.84	2.4
51	Circ. water temp. rise, °F	(From Sanitary System)			4.66	4.78
52	Circulating Pump - Power input, H.P.	2.04	1.91	2.13	8.8	8.4
53	Brine circ. pump-power input, H.P.					

- SPECIFICATIONS -

Forward Plant

The forward plant consists of two Brunswick-Ermeskeil 6-ton, 4½"x8" vertical, single cylinder, double acting, motor driven CO_2 compressors; one shell and coil type brine cooler (8 sections of 3 coils each); two CO_2 double pipe condensers (8 sections each); two brine circulating pumps and the necessary scale traps, separators, etc.

The circulating water supply for the CO_2 condensers is taken from the ship's sanitary system.

This plant serves the two forward scuttlebutts, a photographic laboratory fresh and salt water cooler, and an air cooler.

One CO_2 compressor, one brine circulating pump, one CO_2 condenser (8 sections) and one brine cooler section (3 coils) should produce concurrently for a period of 24 hrs. the following refrigeration, with circulating water at 88° F.

(a) - Photographic Laboratory Cooler
Cool 4 gal./min. salt water from 88° F. to 68° F. for one-half hour every alternate half hour.
Cool 1 gal./min.fresh water from 88° F. to 68° F. continuously.

(b) - Air Cooler-
Reduce the air temperature through the air cooler to, and maintain at, 68° F. when absorbing 21,000 B.T.U./hr.

(c) - Scuttlebutts -
Cool 666 gals/ 24 hrs. of drinking water (333 gal. per scuttlebutt) from 90° F. to 50°.

After Plant

The after refrigerating plant consists of three York 6-ton, 4½"x8" vertical, twin cylinder, single acting, motor driven CO_2 compressors; one brine cooler (3 sections); one CO_2 condenser (3 sections); one ice making tank (3 sections); two brine circulating pumps; two CO_2 condenser circulating pumps; CO_2 dryer and the necessary traps, separators, etc.

This plant serves the refrigerated compartments on the 2nd, and 3rd decks, the ice making tanks and the three after scuttlebutts.

Any two CO_2 compressors with one CO_2 condenser, circulating pump, one brine circulating pump, two condenser sections and two brine cooler sections should produce concurrently for a period of 24 hrs., the following refrigeration with circulating water at 85 F., and outside air at 100 F.

(a) - Maintain the refrigerated compartments at the temperatures listed below.

Compartment	Total Volume	Temperature
Meat Rooms	8780 cu. ft.	15 F.
Vegetables	6854 cu. ft.	40 F.
Butter & Eggs	416 cu. ft.	40 F.
Crew's Cooling Rooms	1247 cu. ft.	40 F.
Officers' Cooling Rooms	2336 cu. ft.	40 F.

(b) - Cool 1000 gals. of drinking water per 24 hr. (333 gals. for each scuttlebutt) from 90° F. to 50° F.

(c) - Make 1600 lbs. of ice per 24 hrs.

TABLE II

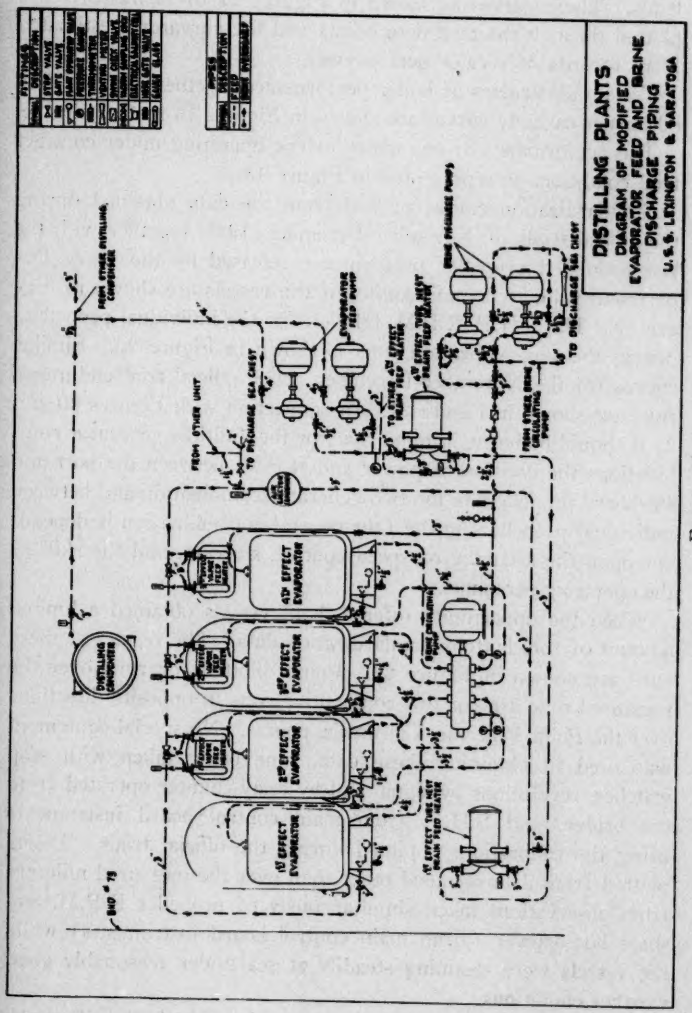

DISTILLING PLANTS
DIAGRAM OF MODIFIED
EVAPORATOR FEED AND BRINE
DISCHARGE PIPING
U.S.S. LEXINGTON & SARATOGA
1ST EFFECT EVAPORATOR
2ND EFFECT EVAPORATOR
3RD EFFECT EVAPORATOR
4TH EFFECT EVAPORATOR
DISTILLING CONDENSER
FRESH ETHER UNIT
FRESH ETHER DISTILLING CONDENSER
TO DISCHARGE SEA CHEST
EVAPORATOR FEED PUMP
BRINE DISCHARGE PUMPS
FRESH ETHER BRINE CIRCULATING PUMP
1ST EFFECT TUBE HEAT MAIN FEED HEATER
BHD "79"

FIGURE 10.

trials. These curves as shown in Figures 11 to 14 inclusive are plotted through the trial data points and the curvatures estimated from records of service performance.

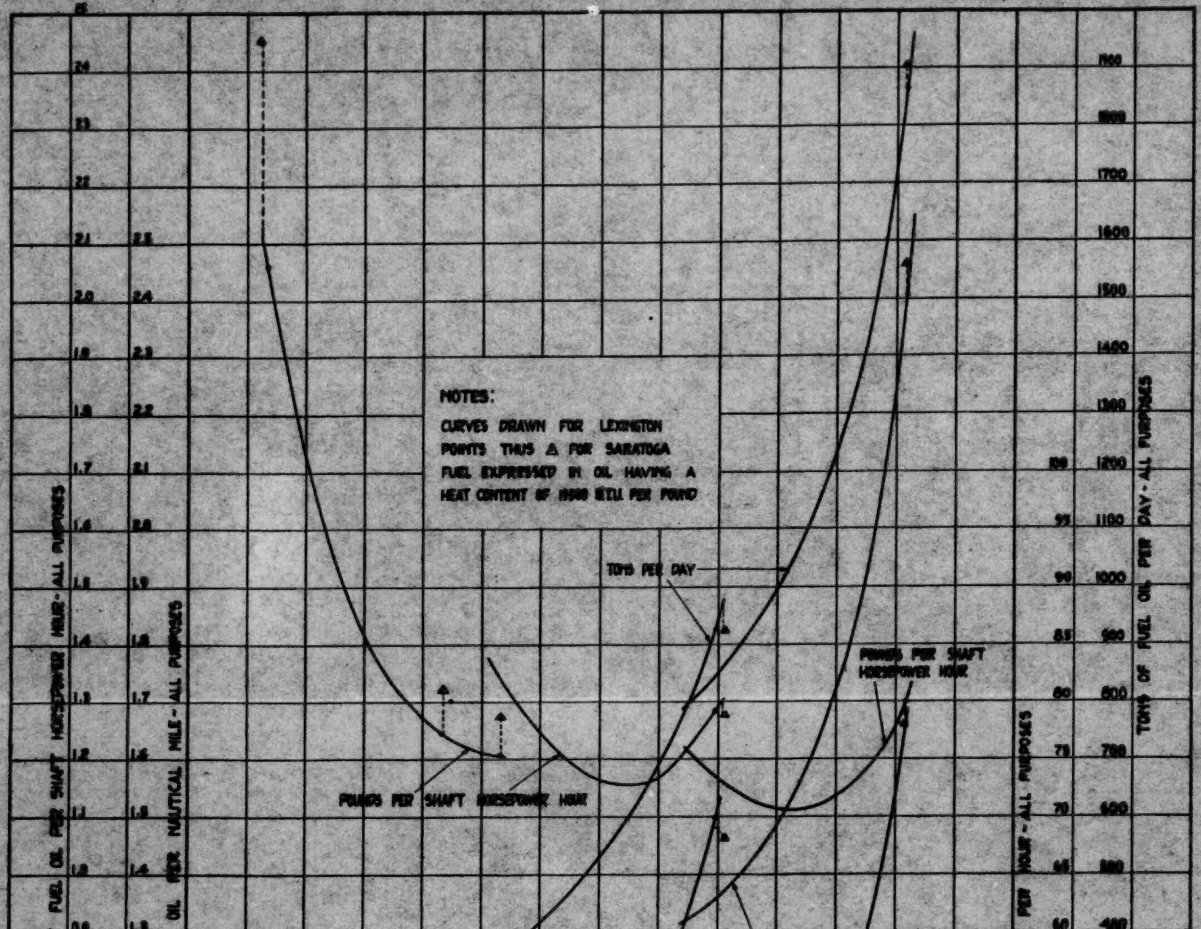
Certain particulars of boiler performance together with the fuel oil burner capacity curves are shown in Figures 15 to 17 inclusive.

The performance of one main turbine operating under contract trial conditions is represented in Figure 18.

Standardization curves plotted from the data obtained during the official trials of November-December, 1928, together with the last estimated propeller performance released by the Navy Department prior to commissioning of the vessels are shown in Figure 19. The S.H.P.-R.P.M. relation for the individual propellers during the standardization runs is shown in Figure 20. Similar curves for the November-December, 1928, official trial endurance runs are shown in Figure 21. In connection with Figures 20 and 21 it should be borne in mind that for the multiple generator combinations the division of power and R.P.M. between the port and starboard propellers in the two-generator combination and between individual propellers in the four-generator combination is dependent upon the accuracy of speed control, steering, and the skill of the operating personnel.

When the opportunity offered, both vessels obtained a limited amount of foul bottom standardization data. The results of these runs are shown in Figure 22. Points plotted for runs over the measured mile are for two consecutive runs in opposite directions over the Point Vincente, California, course. No special equipment was used in obtaining these data, time being taken with stop watches, revolutions by signal and by relay counter operated from the bridge, and S.H.P. from main control board instruments using the calibrations obtained during the official trials. Points plotted from data obtained other than over the measured mile are from observations taken simultaneously of propeller R.P.M. and shaft horsepower (from main control board instruments) while the vessels were steaming steadily at sea under reasonably good weather conditions.

In accordance with usual Naval practice practically all of the auxiliary machinery with the exception of the heat transfer apparatus underwent tests to determine operating characteristics at the


NOTES:
CURVES DRAWN FOR LEXINGTON
POINTS THUS △ FOR SARATOGA
FUEL EXPRESSED IN OIL HAVING A
HEAT CONTENT OF 18500 B.T.U. PER POUND
TONS PER DAY
POUNDS PER SHAFT HORSEPOWER HOUR
POUNDS PER SHAFT HORSEPOWER HOUR
FUEL OIL PER SHAFT HORSEPOWER HOUR - ALL PURPOSES
OIL PER NAUTICAL MILE - ALL PURPOSES
TONS PER HOUR - ALL PURPOSES
TONS OF FUEL OIL PER DAY - ALL PURPOSES

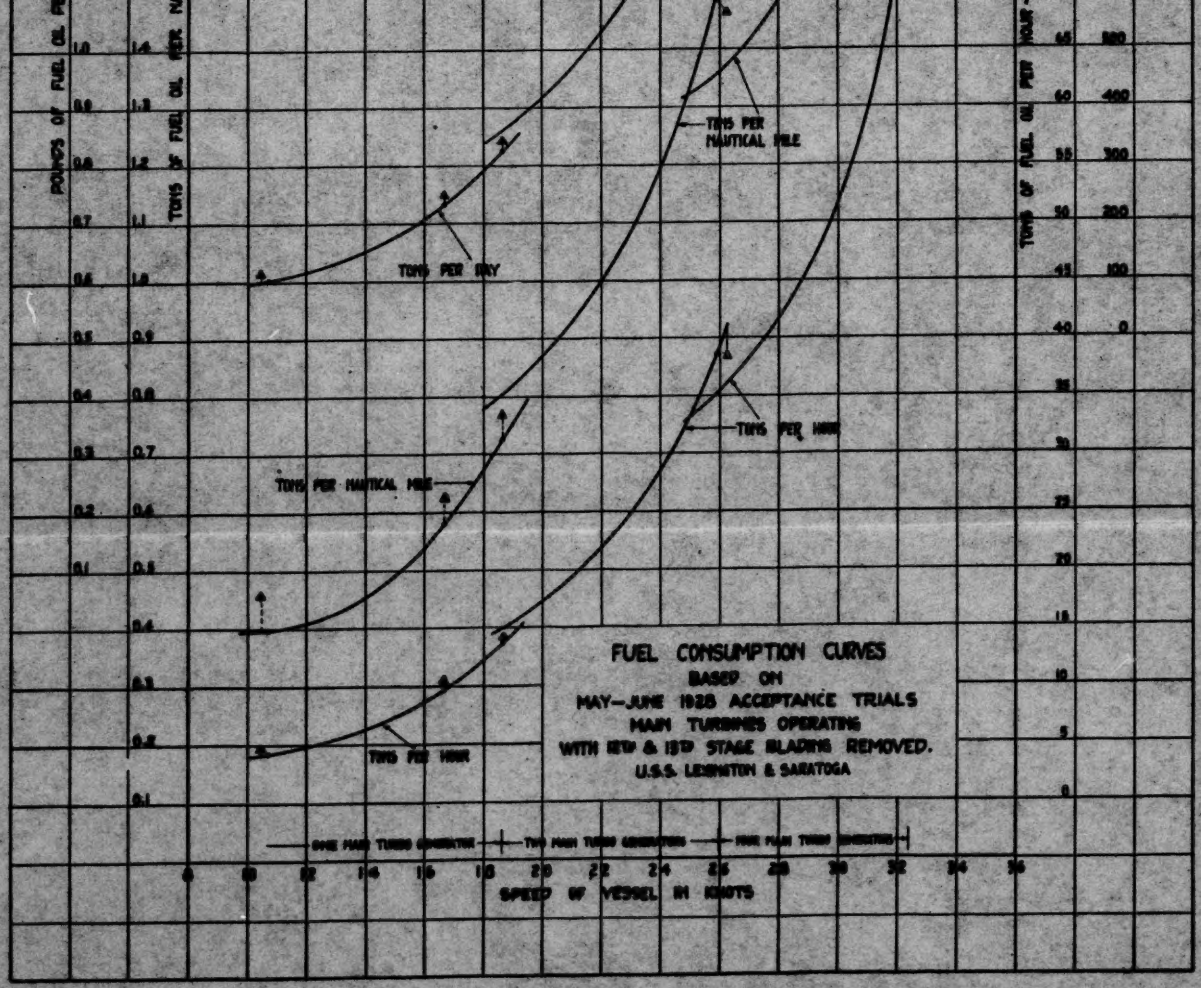
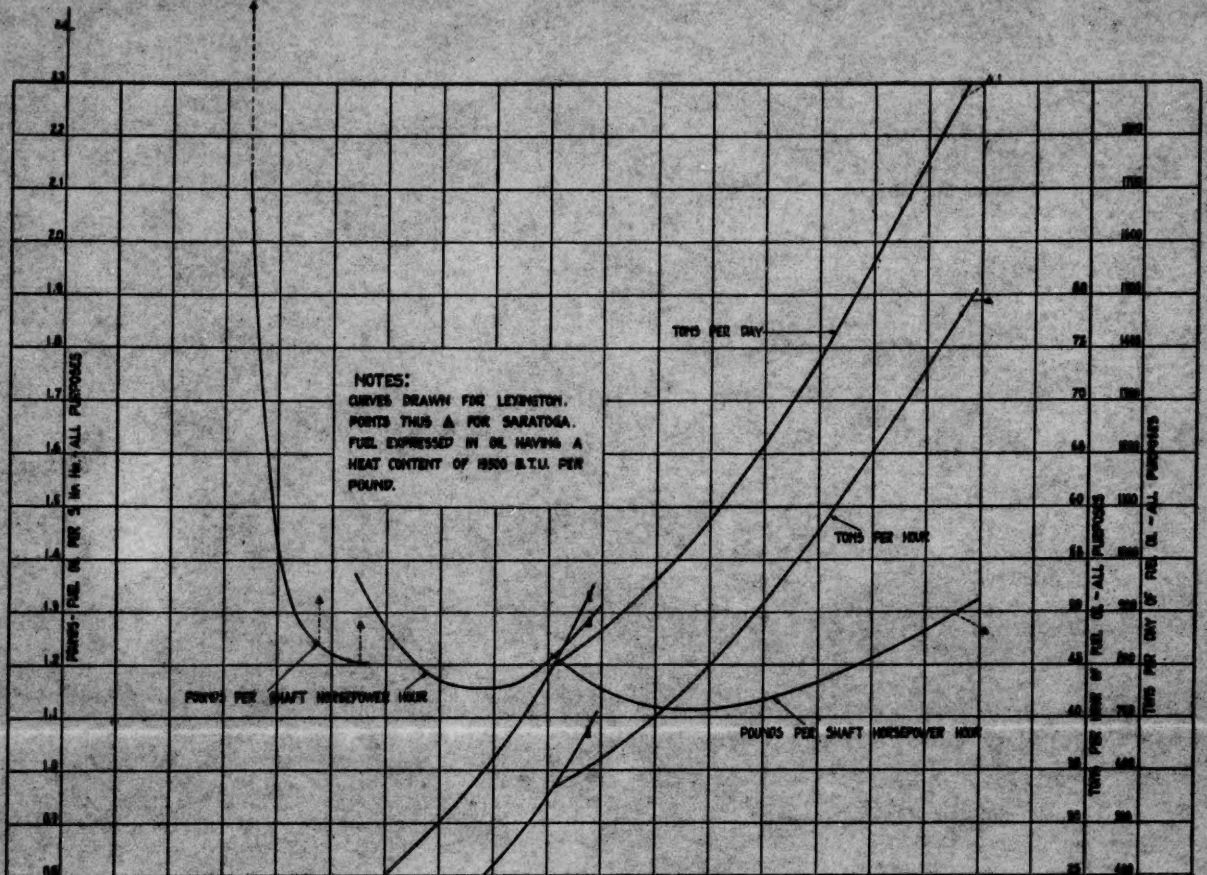

POUNDS OF FUEL OIL PER
TONS OF FUEL OIL PER NAU
TONS OF FUEL OIL PER HOUR
TONS PER DAY
TONS PER NAUTICAL MILE
TONS PER HOUR
FUEL CONSUMPTION CURVES
BASED ON
MAY–JUNE 1928 ACCEPTANCE TRIALS
MAIN TURBINES OPERATING
WITH 1ST & 1ST STAGE BLADING REMOVED.
U.S.S. LEXINGTON & SARATOGA
SPEED OF VESSEL IN KNOTS

FIGURE 11.


NOTES:
CURVES DRAWN FOR LEXINGTON.
POINTS THUS Δ FOR SARATOGA.
FUEL EXPRESSED IN OIL HAVING A HEAT CONTENT OF 18500 B.T.U. PER POUND.
TONS PER DAY
TONS PER HOUR
POUNDS PER SHAFT HORSEPOWER HOUR
POUNDS PER SHAFT HORSEPOWER HOUR
POUNDS OF FUEL OIL PER S.H.P. - ALL PURPOSES
TONS PER DAY OF FUEL OIL - ALL PURPOSES

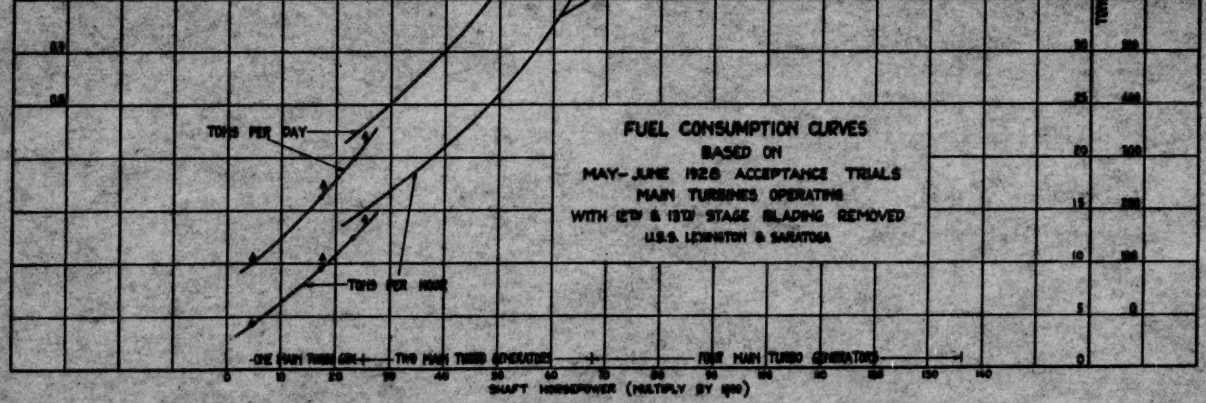
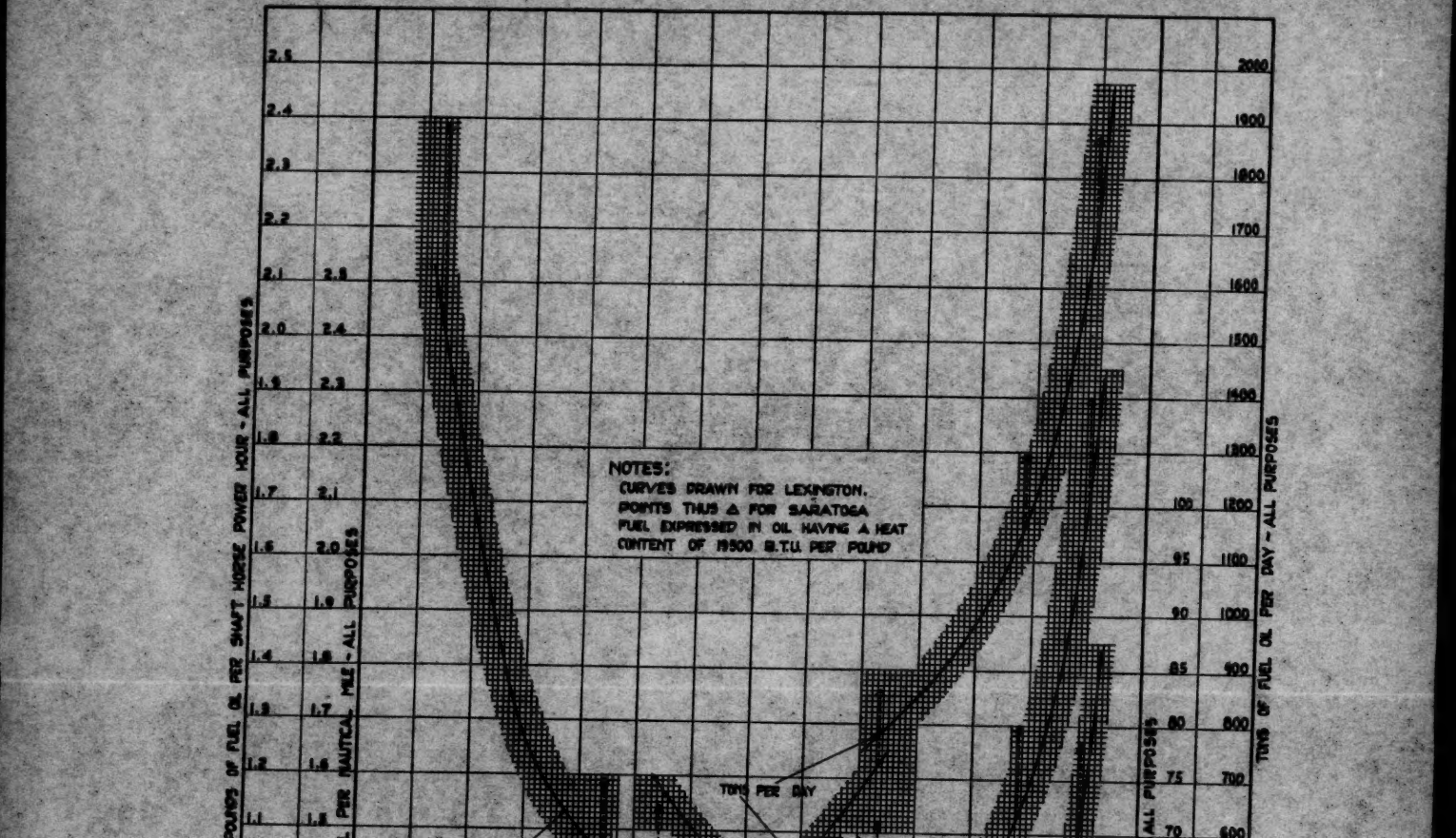

FUEL CONSUMPTION CURVES
BASED ON
MAY-JUNE 1928 ACCEPTANCE TRIALS
MAIN TURBINES OPERATING
WITH 12TH & 13TH STAGE BLADING REMOVED
U.S.S. LEXINGTON & SARATOGA
TONS PER DAY
TONS PER HOUR
ONE MAIN TURBO GEN.
TWO MAIN TURBO GENERATORS
FOUR MAIN TURBO GENERATORS
SHAFT HORSEPOWER (MULTIPLY BY 100)

FIGURE 12


NOTES:
CURVES DRAWN FOR LEXINGTON.
POINTS THUS △ FOR SARATOGA
FUEL EXPRESSED IN OIL HAVING A HEAT
CONTENT OF 19500 B.T.U. PER POUND
POUNDS OF FUEL OIL PER SHAFT HORSE POWER HOUR - ALL PURPOSES
POUNDS OF FUEL OIL PER NAUTICAL MILE - ALL PURPOSES
TONS PER DAY
TONS OF FUEL OIL PER DAY - ALL PURPOSES
2.5
2.4
2.3
2.2
2.1
2.0
1.9
1.8
1.7
1.6
1.5
1.4
1.3
1.2
1.1
2.8
2.4
2.3
2.2
2.1
2.0
1.9
1.8
1.7
1.6
1.5
2000
1900
1800
1700
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85
80
75
70

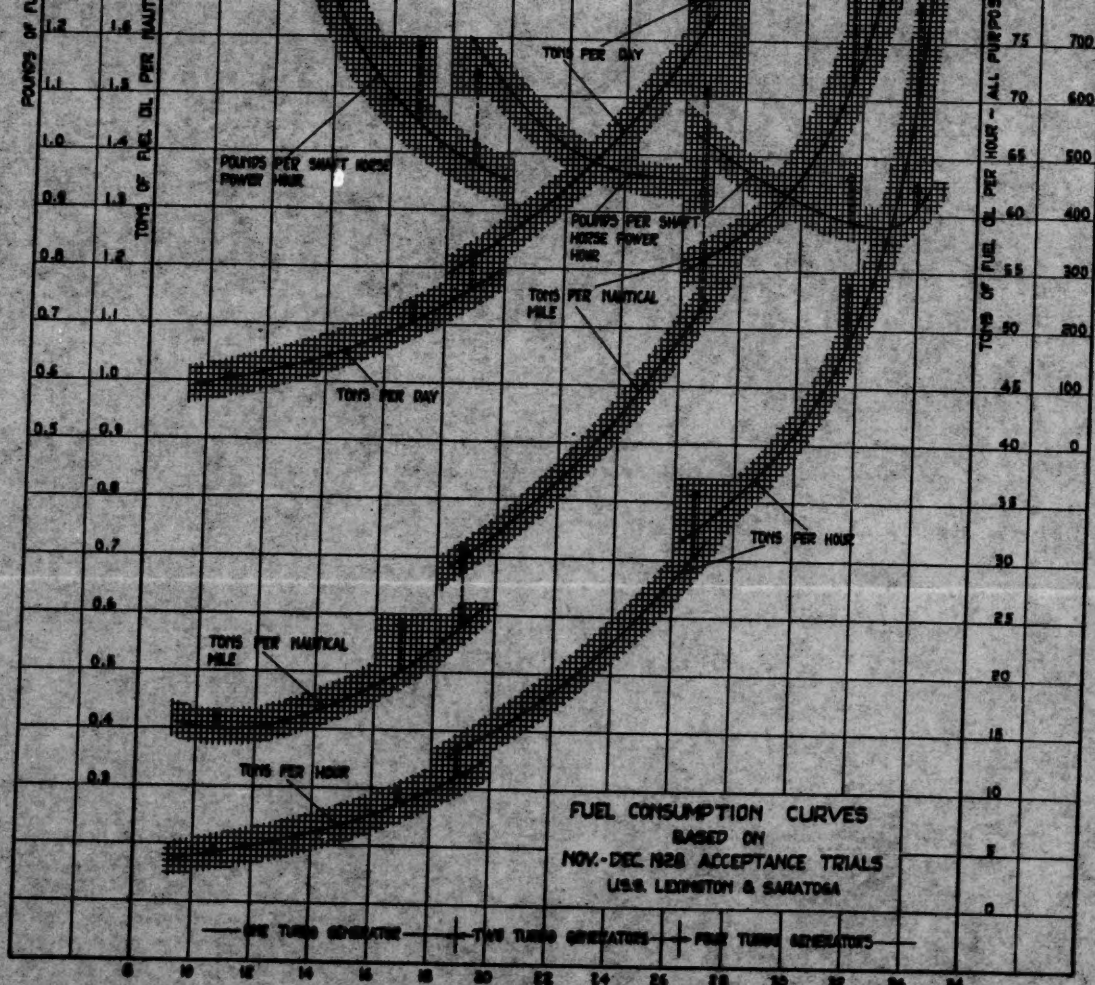

POUNDS OF FUEL OIL PER SHAFT HORSE POWER HOUR
TONS OF FUEL OIL PER NAUTICAL MILE
TONS OF FUEL OIL PER HOUR - ALL PURPOSES
TONS PER DAY
POUNDS PER SHAFT HORSE POWER HOUR
TONS PER NAUTICAL MILE
TONS PER HOUR
FUEL CONSUMPTION CURVES
BASED ON
NOV.-DEC. 1928 ACCEPTANCE TRIALS
U.S.S. LEXINGTON & SARATOGA
ONE TURBO GENERATOR
TWO TURBO GENERATORS
FOUR TURBO GENERATORS
SPEED OF VESSEL IN KNOTS

FIGURE 13.

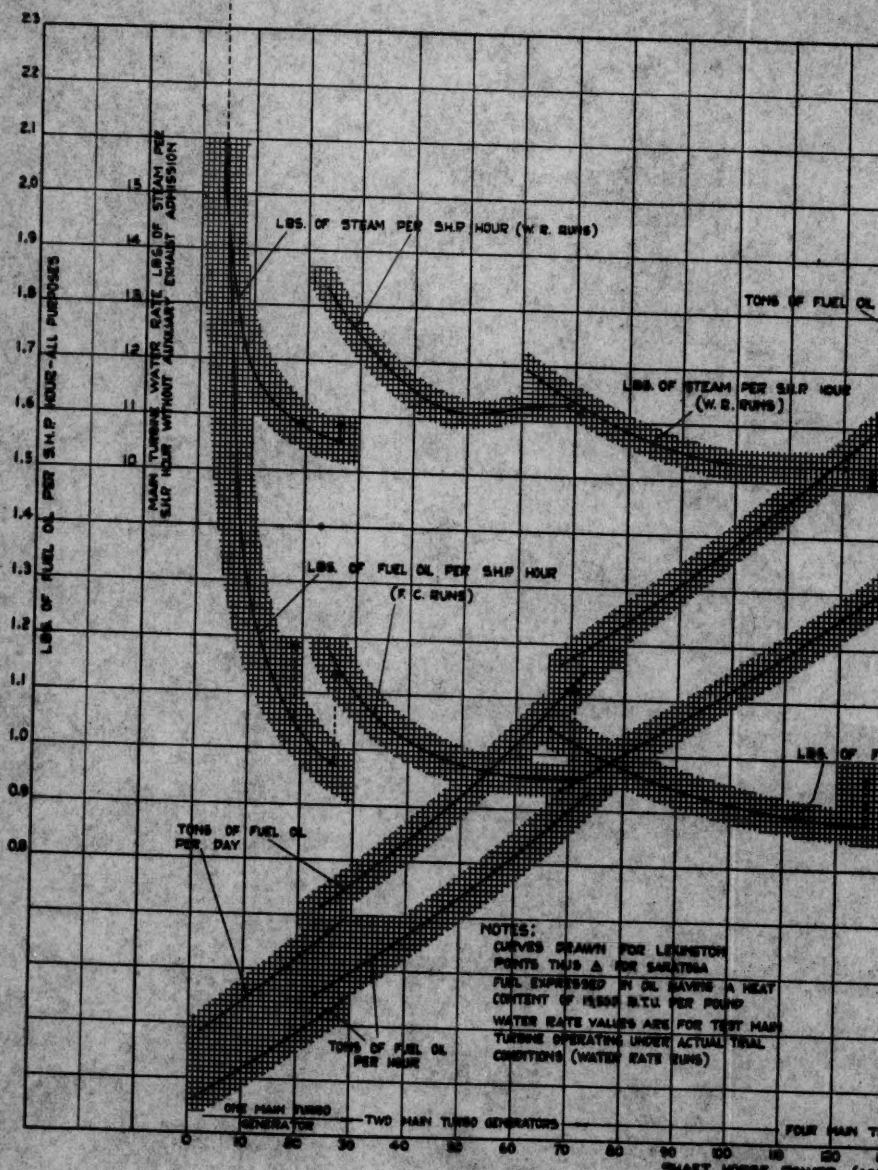
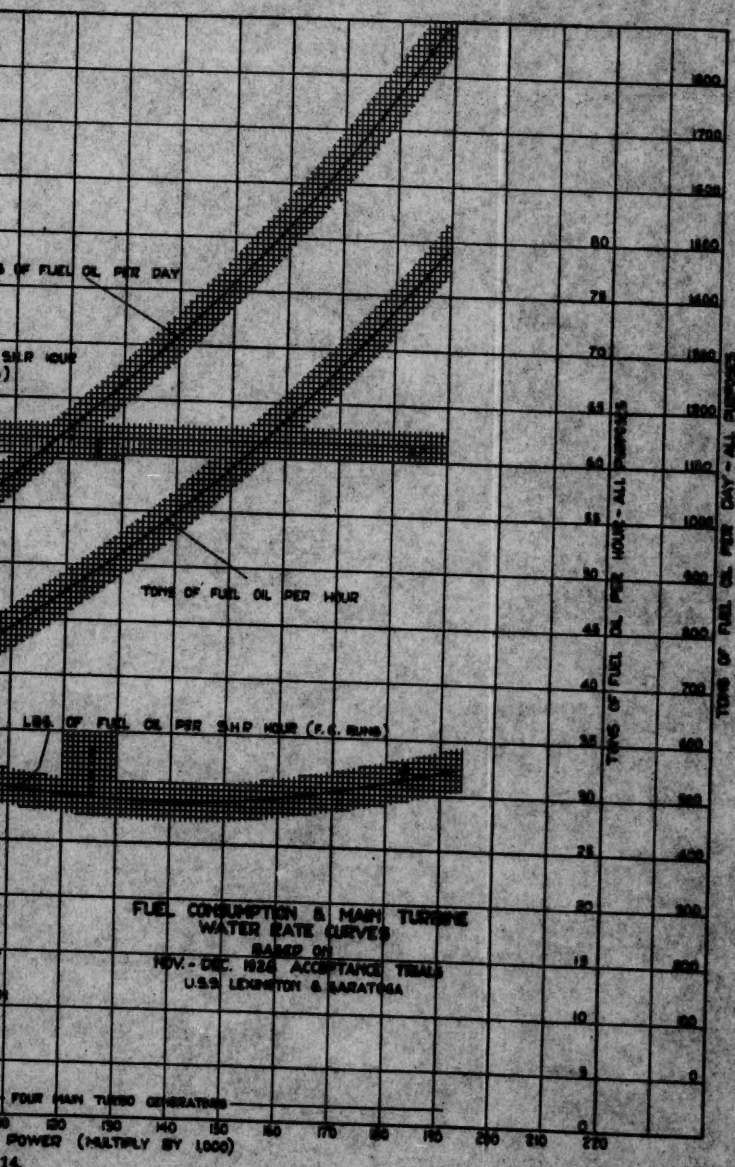

LBS. OF FUEL OIL PER S.H.P HOUR-ALL PURPOSES
MAIN TURBINE WATER RATE LBS. OF STEAM PER S.H.P HOUR WITHOUT AUXILIARY EXHAUST ADMISSION
LBS. OF STEAM PER S.H.P HOUR (W. R. RUNS)
TONS OF FUEL OIL
LBS. OF STEAM PER S.H.P HOUR (W. R. RUNS)
LBS. OF FUEL OIL PER S.H.P HOUR (F. C. RUNS)
TONS OF FUEL OIL PER DAY
TONS OF FUEL OIL PER HOUR
NOTES:
CURVES DRAWN FOR LEXINGTON
POINTS THUS Δ FOR SARATOGA
FUEL EXPRESSED IN OIL HAVING A HEAT CONTENT OF 19500 B.T.U. PER POUND
WATER RATE VALUES ARE FOR TEST MAIN TURBINE OPERATING UNDER ACTUAL TRIAL CONDITIONS (WATER RATE RUNS)
ONE MAIN TURBO GENERATOR
TWO MAIN TURBO GENERATORS
FOUR MAIN T
SHAFT HORSE POWER (M

FIGURE 14.


TONS OF FUEL OIL PER DAY
TONS OF FUEL OIL PER HOUR - ALL PURPOSES
TONS OF FUEL OIL PER DAY - ALL PURPOSES
TONS OF FUEL OIL PER HOUR
LBS. OF FUEL OIL PER S.H.P. HOUR (F.C. RUNS)
FUEL CONSUMPTION & MAIN TURBINE WATER RATE CURVES
BASED ON
NOV. - DEC. 1926 ACCEPTANCE TRIALS
U.S.S. LEXINGTON & SARATOGA
FOUR MAIN TURBO GENERATORS
POWER (MULTIPLY BY 1000)

14.

works of the manufacturers or at the Engineering Experiment Station at Annapolis. The most important of the data thus obtained for the *Lexington* and *Saratoga* are graphically presented in Figures 23 to 68 inclusive, alphabetically indexed in Table 12.

TABLE 12 Sheet 1.

INDEX OF CURVE SHEETS FOR PERFORMANCE OF AUXILIARY MACHINERY USS LEXINGTON AND SARATOGA

FIGURE	UNIT	TITLE	SHIP
23	Blowers, forced draft	Blower Performance, Capacity vs. H.p.m.	Lexington and Saratoga
24	" "	Blower Performance, Capacity vs Static Head (Const. speed).	" "
25	" "	Blower Performance, Capacity vs. B.H.P.	" "
26	" "	Blower Performance, Characteristic curve of fan.	" "
27	" "	Turbine Performance, Nozzle Pressure vs. S.h.p., 2 and 6 Nozzles.	"
28	" "	Turbine Performance, Nozzle Pressure vs. S.h.p. 4 nozzles.	" "
29	" "	Steam Consumption.	" "
30	Electric Plant, D.C.	Total D.C. Power and D.C. Power for Excitation, Underway.	"
31	" "	Total D.C. Load Curves, at Anchor.	Lexington
32	Generators, 750 kw. turbine driven.	Parallel Operating Characteristics.	Lexington and Saratoga
33	" "	Load vs. Water Rate	" "
34	" "	Water Rate Correction Curves.	" "
35	Heaters, fuel oil service.	Service Performance Characteristics, Heat transfer coefficients.	Lexington

TABLE 12 (Cont'd.) Sheet 2

FIGURE	UNIT	TITLE	SHIP
36	Heaters, main feed water.	Service Performance Characteristics, Heat transfer coefficients.	Lexington
37	Pumps, auxiliary and reserve feed.	Steam Consumption, 350# discharge.	Lexington and Saratoga
38	" "	" " 100# "	" "
39	Pumps, distiller circulating (MD)	Pump Characteristics.	" "
40	Pumps, distiller condensate (MD)	" "	" "
41	Pumps, distiller, fresh water (MD)	" "	" "
42	Pumps, dynamo condenser air.	Capacity Curves.	" "
43	Pumps, dynamo condenser circulating.	Pump Characteristics.	" "
44	Pumps, dynamo condenser circulating.	Steam Consumption.	" "
45	Pumps, dynamo condenser condensate.	Pump Characteristics and Steam Consumption.	" "
46	Pumps, evaporator brine circulating (MD).	Pump Characteristics.	" "
47	Pumps. evaporator brine discharge (MD).	" "	" "
48	Pumps, evaporator feed (MD)	" "	" "
49	Pumps, fire and bilge, (Centrif.MD).	" "	" "

TABLE 12

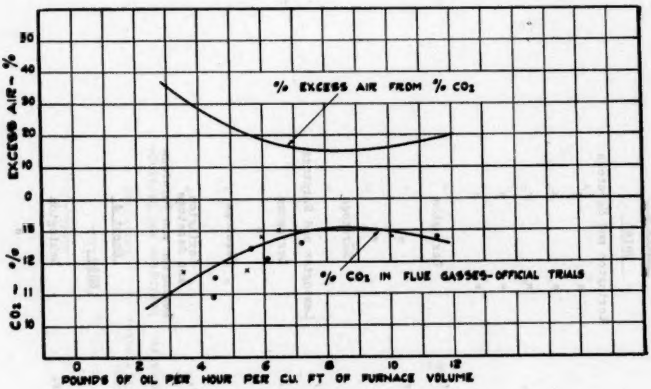
Sheet 3

FIGURE	UNIT	TITLE	SHIP
50	Pumps, fire and bilge(Recip.steam)	Steam Consumption.	Lexington and Saratoga
51	Pumps, fuel oil booster.	" "	" "
52	" " " recirculating.	" "	" "
53	" " " service.	Pump Characteristics.	" "
54	" " " "	Steam Consumption.	" "
55	" main condenser air.	Capacity Curves.	" "
56	" " " circulating.	Pump Characteristics, Constant Head.	Lexington
57	" " " "	Pump Characteristics, Variable Head.	"
58	" " " "	Steam Consumption.	"
59	" " " "	" "	Saratoga
60	" " " condensate	Pump Characteristics, Steam Consumption.	Lexington and Saratoga
61	" " feed	Pump Characteristics.	Lexington
62	" " "	Steam Consumption.	"
63	" " "	" "	Saratoga
64	" main generator air cooler and hydraulic ejector supply.	Pump Characteristics.	Lexington and Saratoga
65	ditto	Steam Consumption.	Lexington and Saratoga

TABLE 12 (Cont'd.)

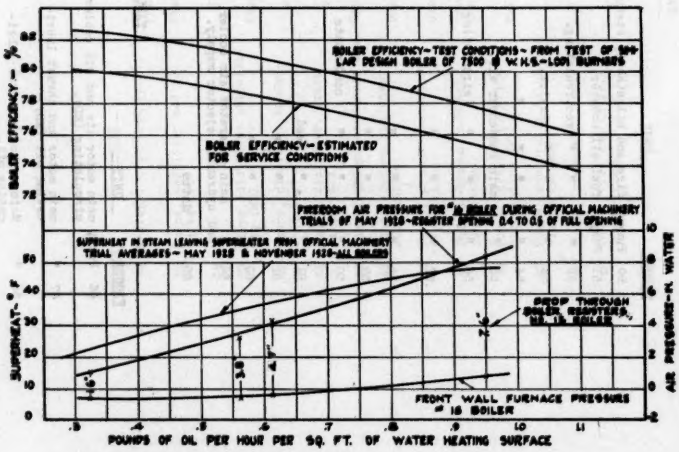
Sheet 4

FIGURE	UNIT	TITLE	SHIP
66	Pumps, main motor air and oil cooler circulating (MD).	Pump Characteristics.	Lexington
67	" main motor and thrust lubricating oil (MD).	" "	"
68	" Main turbo-generator lubricating oil.	" " and water rates.	Lexington & Saratoga



NOTE:-
X - OFFICIAL TRIALS - MAY 1928
O - OFFICIAL TRIALS - NOVEMBER 1928
VALUES PLOTTED ABOVE ARE AVERAGES FOR LAST 2 HOURS OF OFFICIAL MACHINERY TRIALS - NO. 16 BOILER & 1 RUN NO. 10
BOILERS FITTED WITH B & W CO. LODI BURNERS.

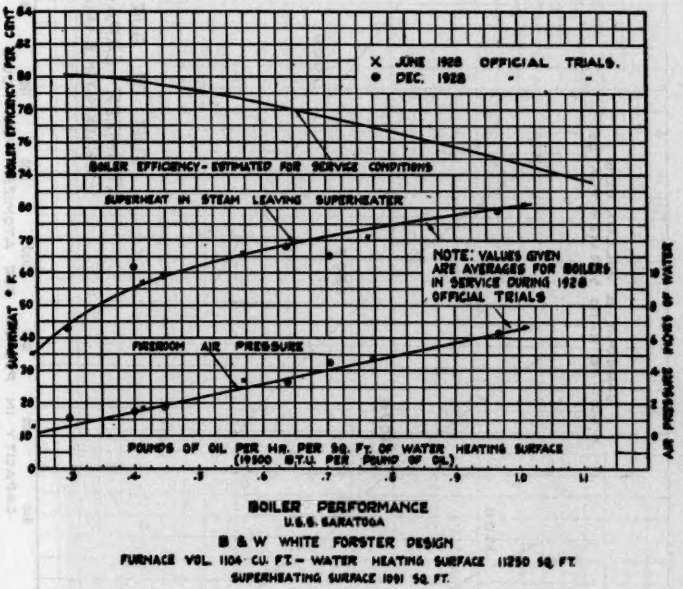
FOR FURTHER PARTICULARS SEE TABULAR DATA FOR OFFICIAL TRIALS.


BOILER EFFICIENCY - %
BOILER EFFICIENCY - TEST CONDITIONS - FROM TEST OF SIMILAR DESIGN BOILER OF 7500 # W.H.S. - LODI BURNERS
BOILER EFFICIENCY - ESTIMATED FOR SERVICE CONDITIONS
FREEBOARD AIR PRESSURE FOR # BOILER DURING OFFICIAL MACHINERY TRIALS OF MAY 1928 - REGISTER SETTING 0.4 TO 0.5 OF FULL OPENING
SUPERHEAT IN STEAM LEAVING SUPERHEATER FROM OFFICIAL MACHINERY TRIAL AVERAGES - MAY 1928 & NOVEMBER 1928 - ALL BOILERS
SUPERHEAT - °F
AIR PRESSURE - IN. WATER
DROP THROUGH BOILER SUPERHEATER NO. 16 BOILER
FRONT WALL FURNACE PRESSURE # 16 BOILER
POUNDS OF OIL PER HOUR PER SQ. FT. OF WATER HEATING SURFACE

NOTE: FOR FURTHER PARTICULARS SEE TABULAR DATA FOR OFFICIAL TRIALS.

BETHLEHEM - YARROW BOILER PERFORMANCE
FURNACE VOLUME 1050 CU. FT. WATER HEATING SURFACE 11250 ☐
SUPERHEATING SURFACE 797 ☐
U.S.S. LEXINGTON

FIGURE 15.


BOILER EFFICIENCY - PER CENT
SUPERHEAT ° F.
AIR PRESSURE INCHES OF WATER
X JUNE 1928 OFFICIAL TRIALS.
● DEC. 1928 " "
BOILER EFFICIENCY - ESTIMATED FOR SERVICE CONDITIONS
SUPERHEAT IN STEAM LEAVING SUPERHEATER
NOTE: VALUES GIVEN ARE AVERAGES FOR BOILERS IN SERVICE DURING 1928 OFFICIAL TRIALS
FIREROOM AIR PRESSURE
POUNDS OF OIL PER HR. PER SQ. FT. OF WATER HEATING SURFACE (19500 B.T.U. PER POUND OF OIL)

BOILER PERFORMANCE
U.S.S. SARATOGA
B & W WHITE FORSTER DESIGN
FURNACE VOL. 1104 CU. FT. – WATER HEATING SURFACE 11250 SQ. FT.
SUPERHEATING SURFACE 1091 SQ. FT.

FIGURE 16.

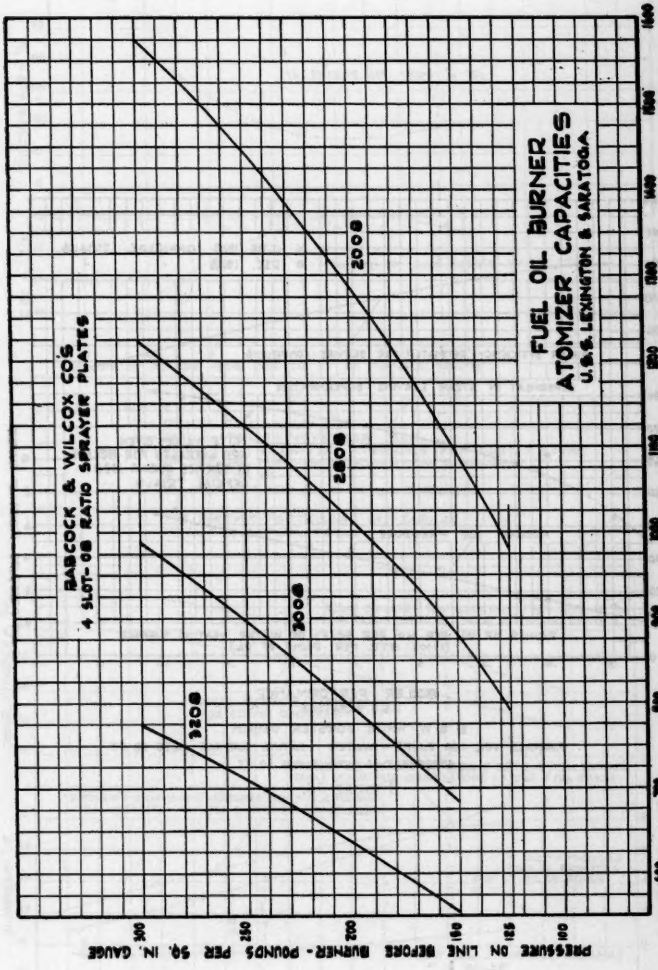

FUEL OIL BURNER
ATOMIZER CAPACITIES
U.S.S. LEXINGTON & SARATOGA
BABCOCK & WILCOX COS
4 SLOT-60 RATIO SPRAYER PLATES
3200
2800
2400
2000
PRESSURE ON LINE BEFORE BURNER - POUNDS PER SQ. IN. GAUGE
CAPACITY IN POUNDS PER ATOMIZER PER HOUR

FIGURE 17.

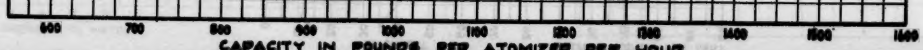

600
700
800
900
1000
1100
1200
1300
1400
1500
1600
CAPACITY IN POUNDS PER ATOMIZER PER HOUR

FIGURE 17.

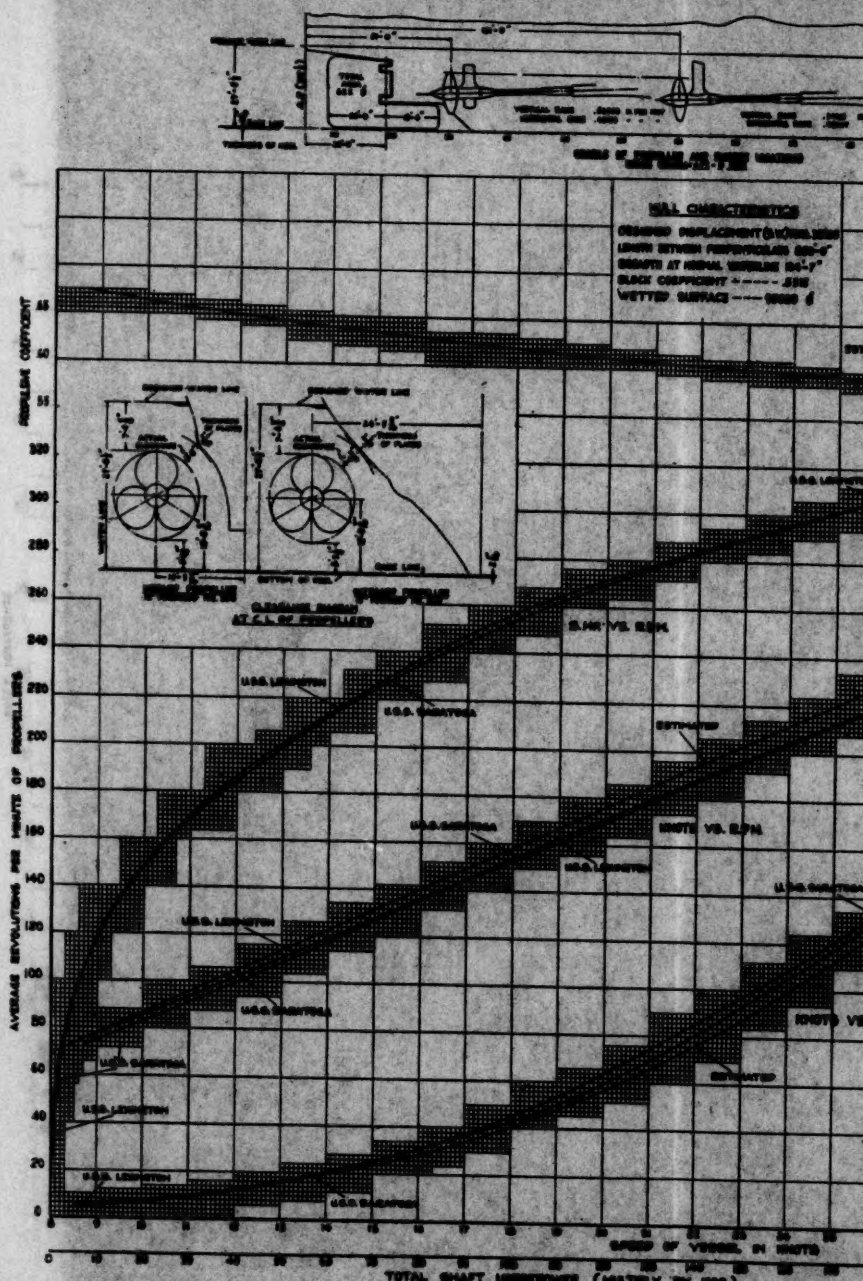
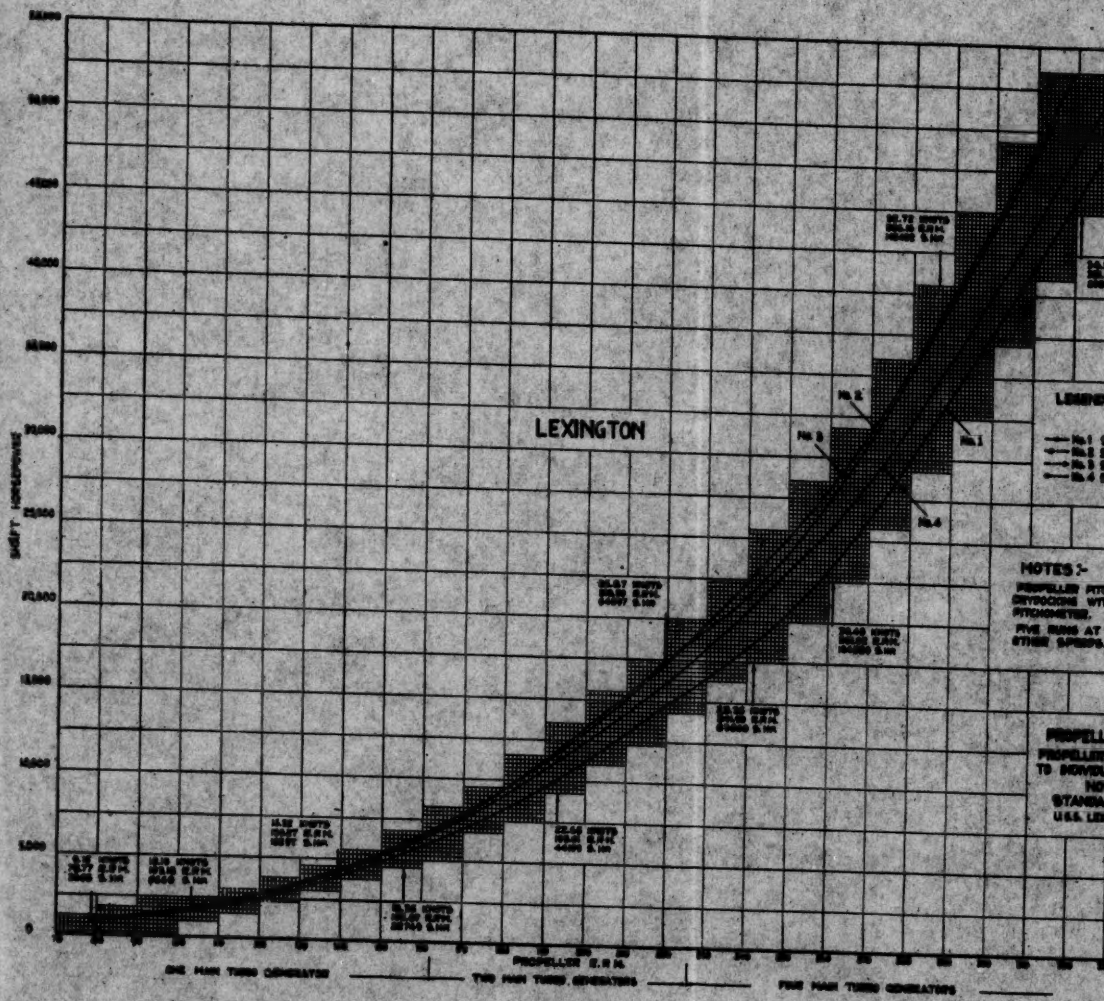

HULL CHARACTERISTICS
AVERAGE REVOLUTIONS PER MINUTE OF PROPELLERS
PROPELLER COEFFICIENT
SPEED OF VESSEL IN KNOTS
TOTAL SHAFT HORSEPOWER (MULTIPLY BY 1000)
U.S.S. LEXINGTON
U.S.S. SARATOGA
ESTIMATED

FIGURE 19.

PROPELLER DATA
NOTE
U.S.S. LEXINGTON & SARATOGA
PROPELLER PERFORMANCE
ESTIMATED & ACTUAL
TOTAL SHAFT HORSEPOWER


LEXINGTON
SHAFT HORSEPOWER
PROPELLER R.P.M.
ONE MAIN TURBO GENERATOR
TWO MAIN TURBO GENERATORS
FIVE MAIN TURBO GENERATORS
No. 2
No. 3
No. 1
No. 4
MOTES:-

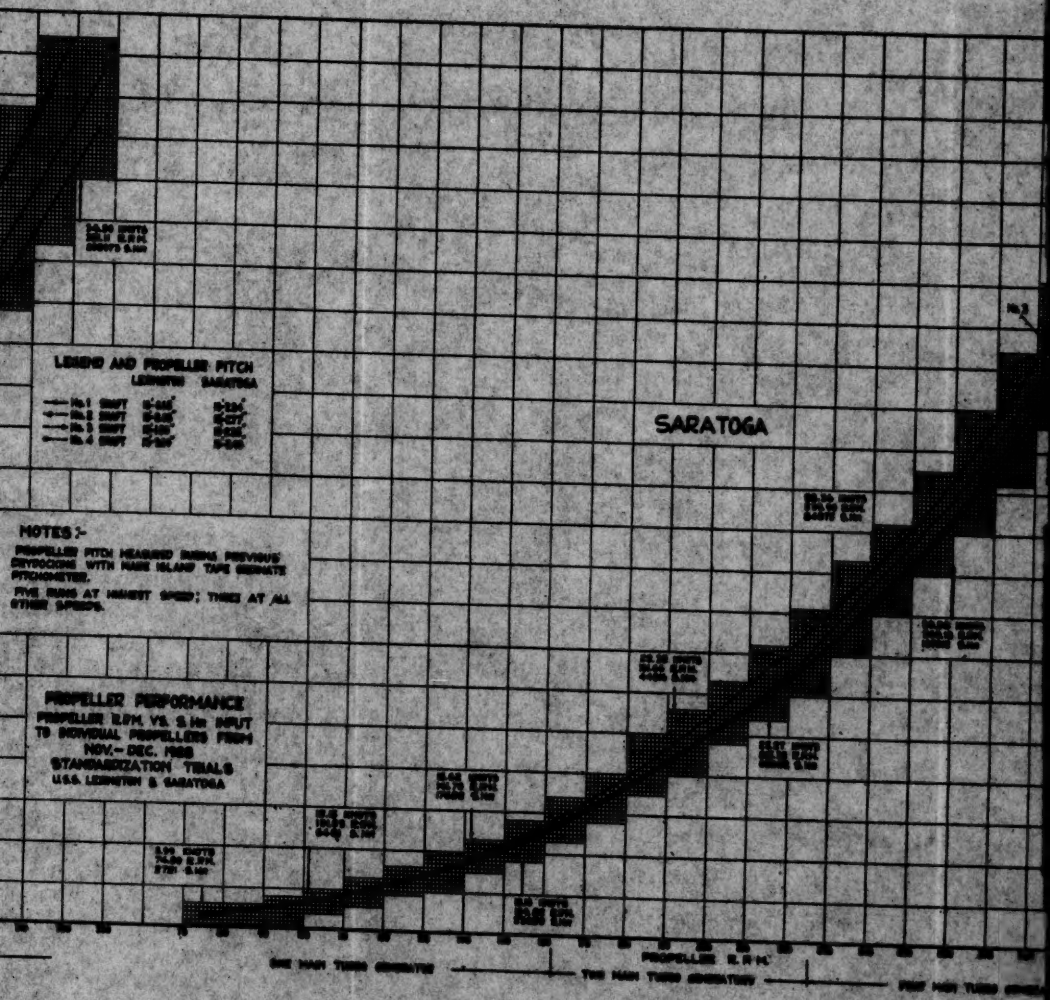

LEGEND AND PROPELLER PITCH
LEXINGTON SARATOGA
No.1 SHAFT
No.2 SHAFT
No.3 SHAFT
No.4 SHAFT
SARATOGA
NOTES:-
PROPELLER PITCH MEASURED DURING PREVIOUS DRYDOCKING WITH HARE ISLAND TAPE SUBMATE PITCHOMETER.
FIVE RUNS AT HIGHEST SPEED; THREE AT ALL OTHER SPEEDS.
PROPELLER PERFORMANCE
PROPELLER R.P.M. VS. S.H.P. INPUT TO INDIVIDUAL PROPELLERS FROM NOV.- DEC. 1925 STANDARDIZATION TRIALS U.S.S. LEXINGTON & SARATOGA
PROPELLER R.P.M.
ONE MAIN TURBO GENERATOR
TWO MAIN TURBO GENERATORS

FIGURE 90.

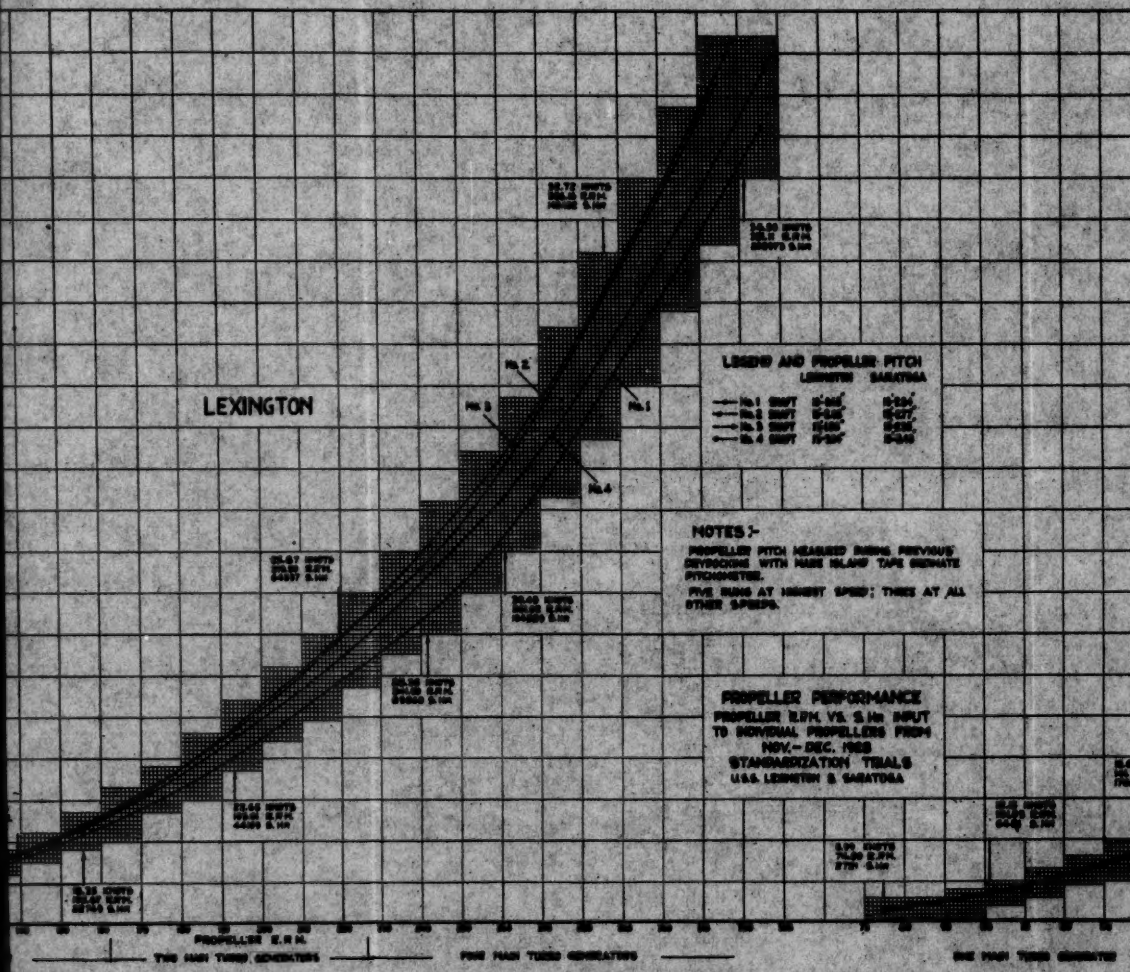
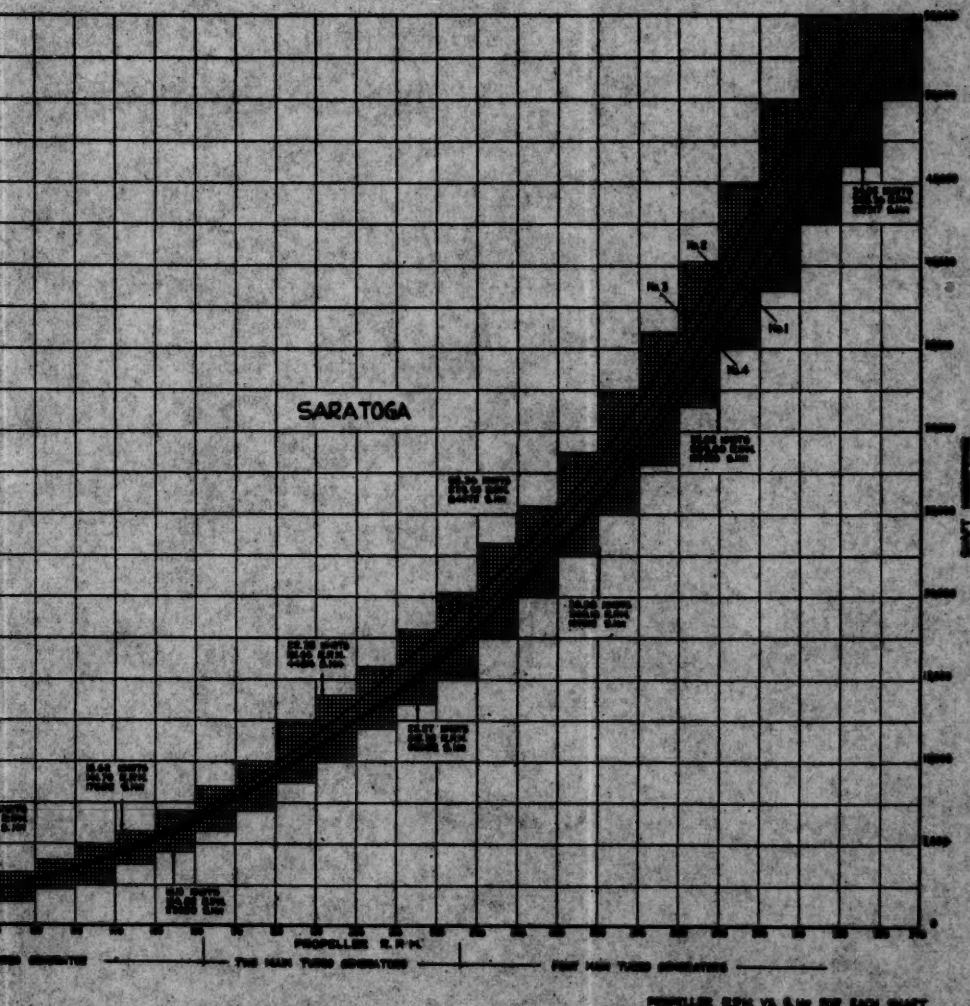
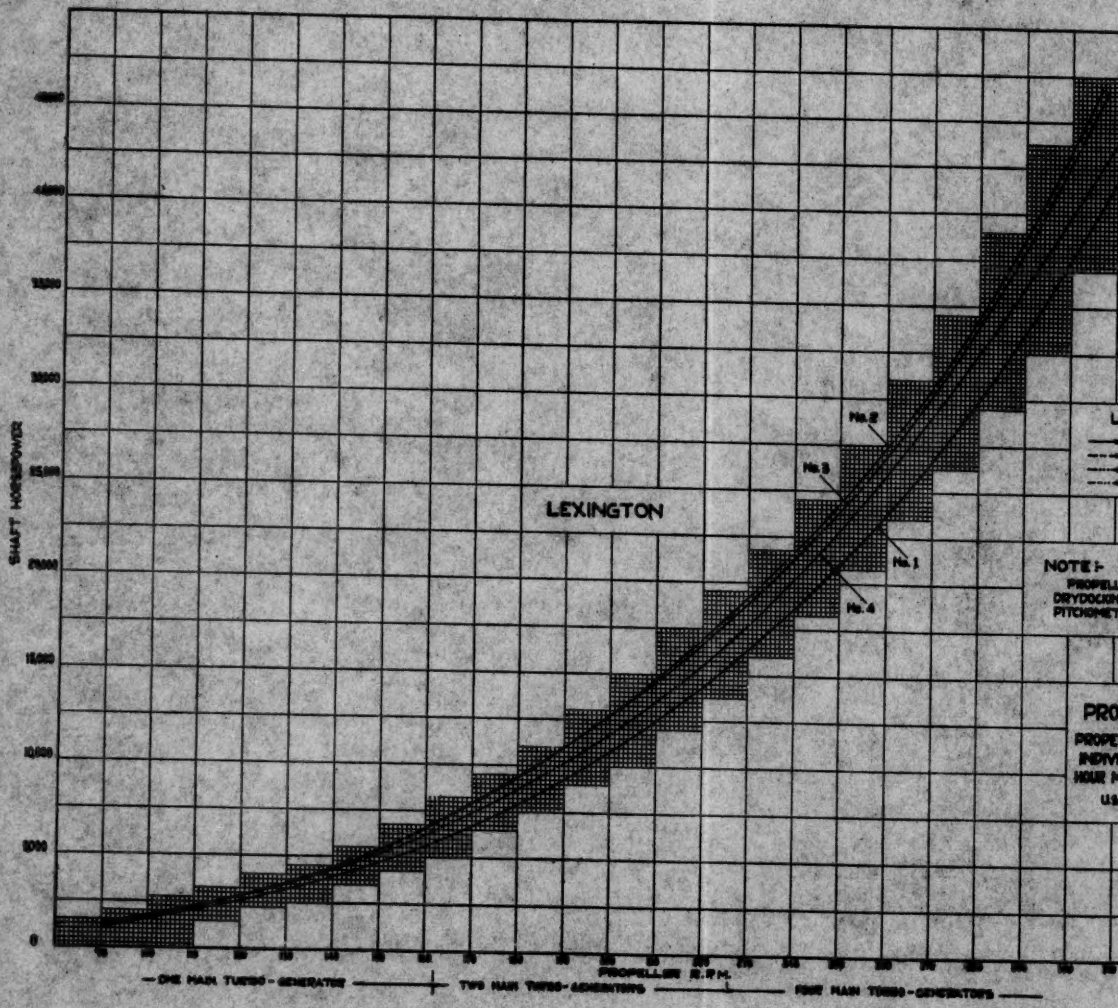

LEXINGTON
No. 2
No. 3
No. 1
No. 4
LEGEND AND PROPELLER PITCH
LEXINGTON SARATOGA
No. 1 SHAFT
No. 2 SHAFT
No. 3 SHAFT
No. 4 SHAFT
NOTES:-
PROPELLER PITCH MEASURED DURING PREVIOUS DRYDOCKING WITH HAGE ISLAND TAPE SUBMATE PITCHOMETER.
FIVE RUNS AT HIGHEST SPEED; THREE AT ALL OTHER SPEEDS.
PROPELLER PERFORMANCE
PROPELLER R.P.M. VS. S.Hp. INPUT TO INDIVIDUAL PROPELLERS FROM NOV.- DEC. 1928 STANDARDIZATION TRIALS
U.S.S. LEXINGTON & SARATOGA
PROPELLER R.P.M.
TWO MAIN TURBO GENERATORS
FIVE MAIN TURBO GENERATORS
ONE MAIN TURBO GENERATOR

FIGURE 20.


SARATOGA


LEXINGTON
No. 2
No. 3
No. 1
No. 4
SHAFT HORSEPOWER
PROPELLER R.P.M.
— ONE MAIN TURBO-GENERATOR
TWO MAIN TURBO-GENERATORS
FOUR MAIN TURBO-GENERATORS

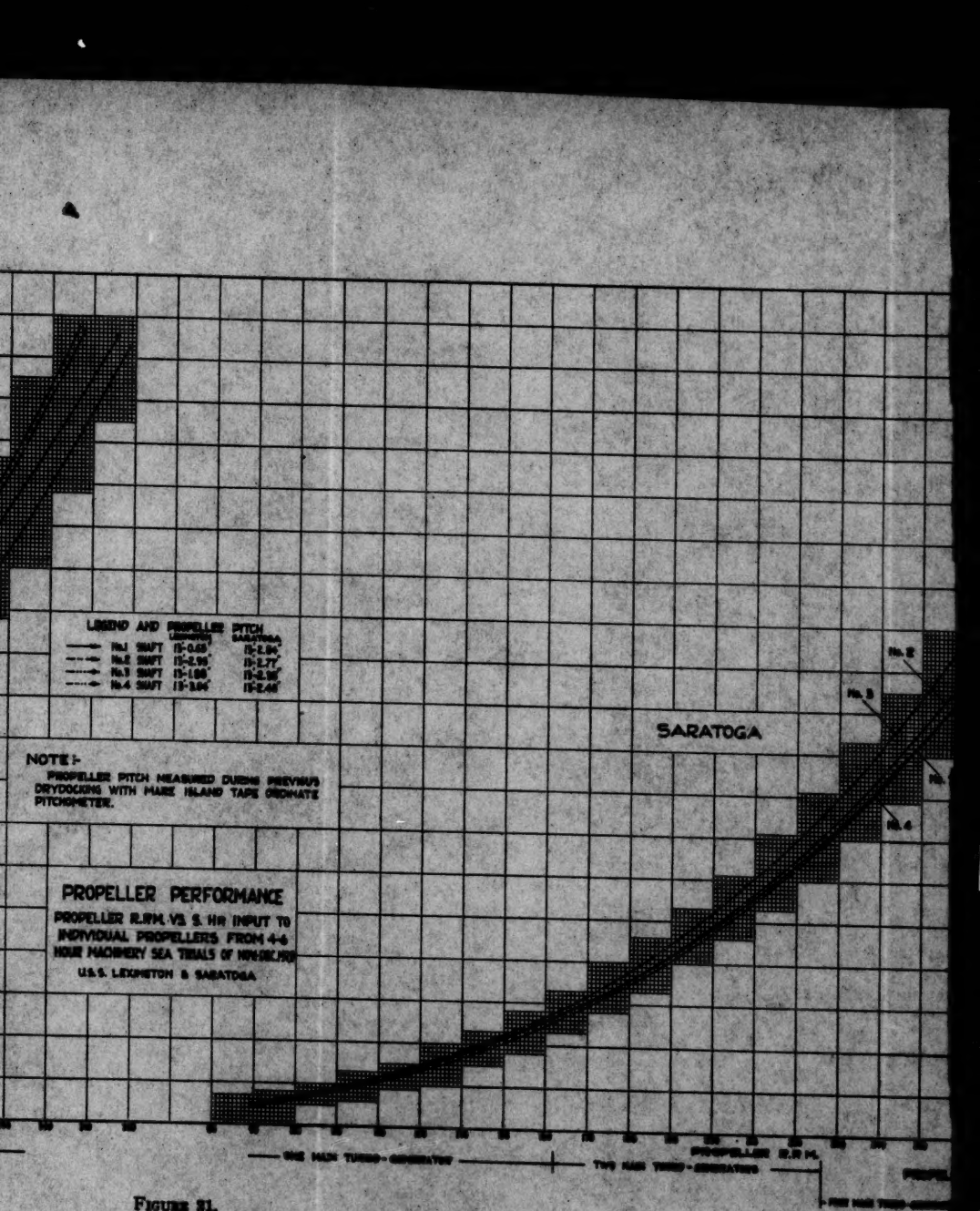

LEGEND AND PROPELLER PITCH
LEXINGTON
SARATOGA
No.1 SHAFT 15'-0.03" 15'-2.84"
No.2 SHAFT 15'-2.99" 15'-2.77"
No.3 SHAFT 15'-1.08" 15'-2.36"
No.4 SHAFT 15'-3.04" 15'-2.46"
NOTE:-
PROPELLER PITCH MEASURED DURING PREVIOUS DRYDOCKING WITH MARE ISLAND TAPE ORDINATE PITCHOMETER.
PROPELLER PERFORMANCE
PROPELLER R.P.M. VS S. HP INPUT TO INDIVIDUAL PROPELLERS FROM 4-6 HOUR MACHINERY SEA TRIALS OF NOVEMBER 1928
U.S.S. LEXINGTON & SARATOGA
SARATOGA
No. 2
No. 3
No. 4
ONE MAIN TURBO-GENERATOR
TWO MAIN TURBO-GENERATORS
PROPELLER R.P.M.

FIGURE 21.

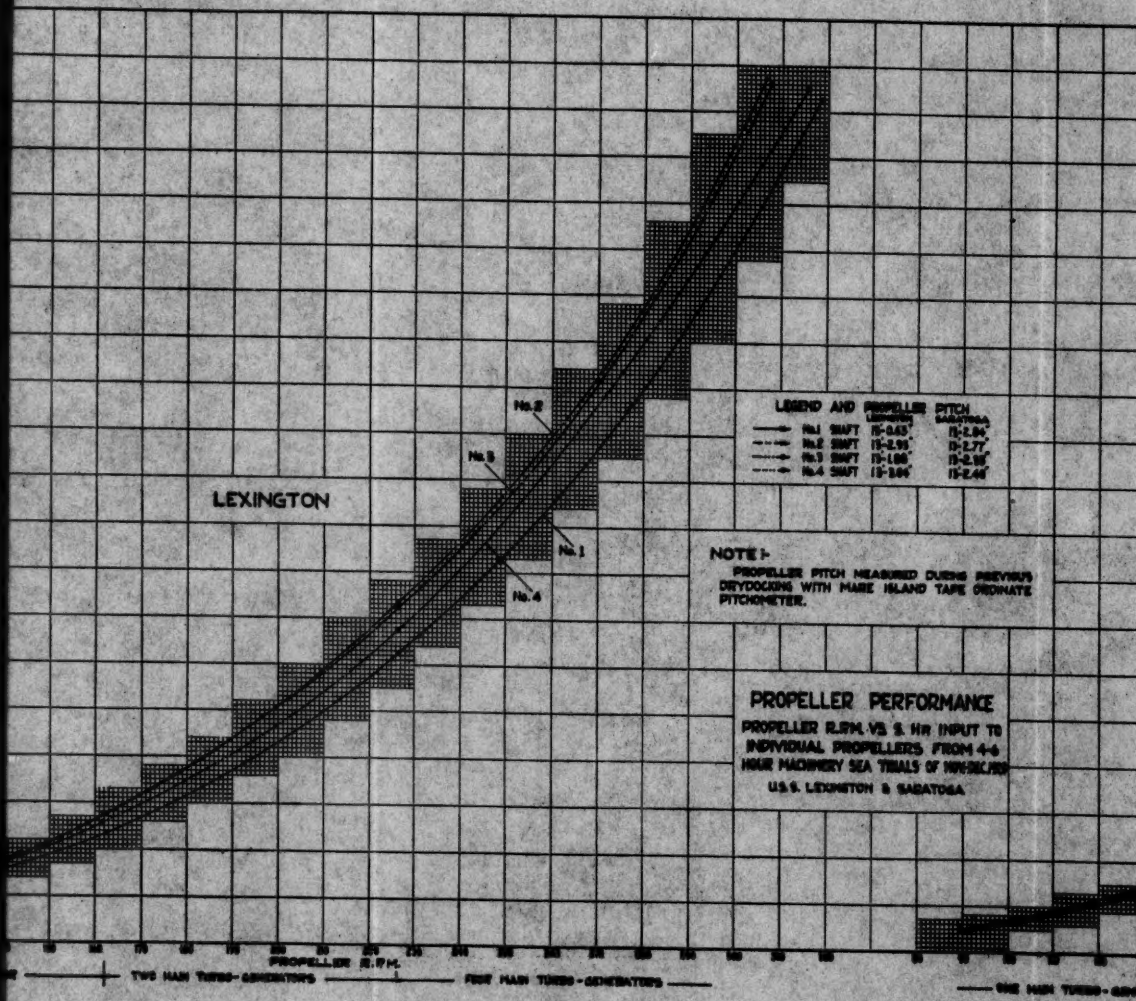
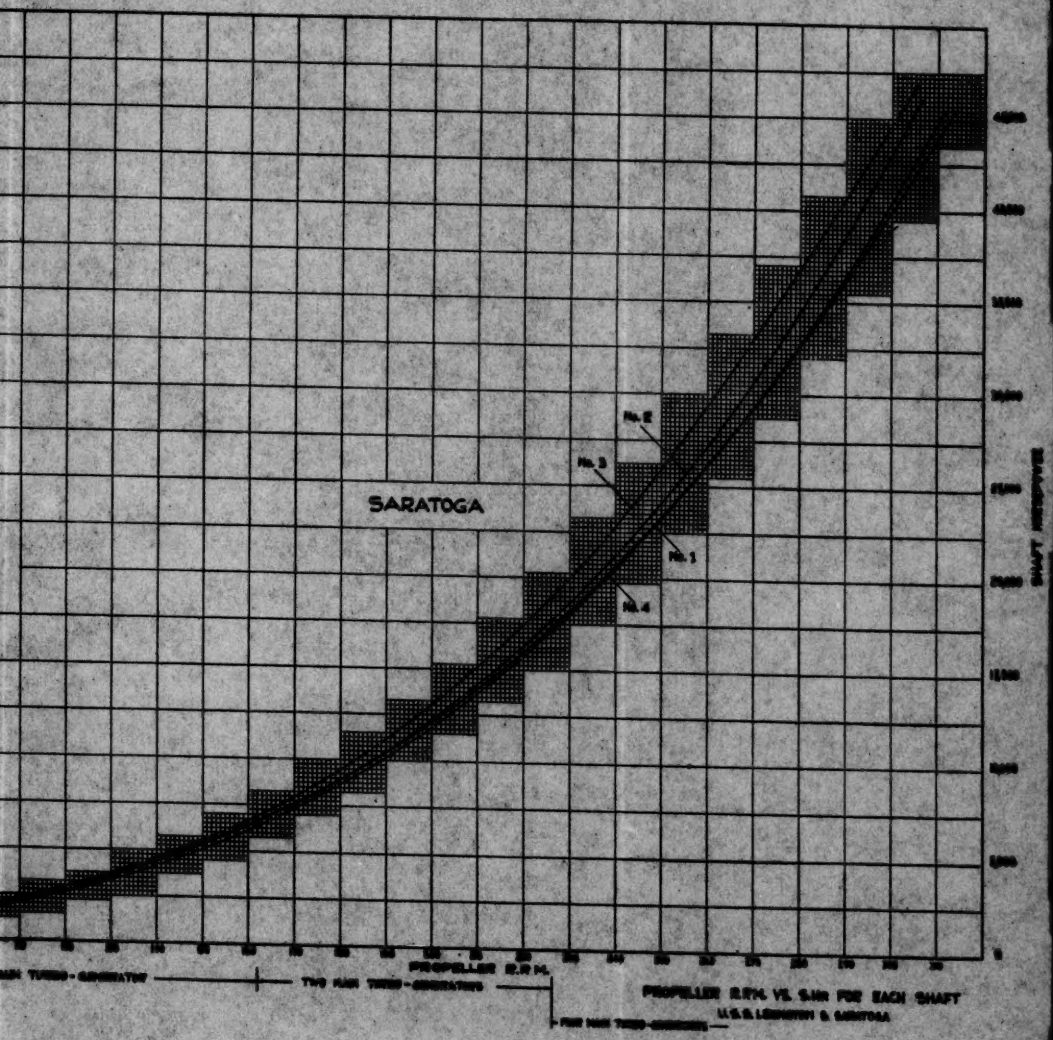

LEXINGTON
No.2
No.3
No.1
No.4
LEGEND AND PROPELLER PITCH
LEXINGTON
SARATOGA
No.1 SHAFT 15-0.63" 15-2.84"
No.2 SHAFT 15-2.99" 15-2.77"
No.3 SHAFT 15-1.05" 15-2.18"
No.4 SHAFT 15-1.84" 15-2.48"
NOTE:-
PROPELLER PITCH MEASURED DURING PREVIOUS DRYDOCKING WITH MARE ISLAND TAPE ORDINATE PITCHOMETER.
PROPELLER PERFORMANCE
PROPELLER R.P.M. VS S. HP INPUT TO INDIVIDUAL PROPELLERS FROM 4-6 HOUR MACHINERY SEA TRIALS OF NOVEMBER 1927
U.S.S. LEXINGTON & SARATOGA
PROPELLER R.P.M.
TWO MAIN TURBO-GENERATORS
FOUR MAIN TURBO-GENERATORS

FIGURE 21.


SARATOGA
No. 2
No. 3
No. 1
No. 4
SHAFT HORSEPOWER
PROPELLER R.P.M.
TWO MAIN TURBO-GENERATORS
PROPELLER R.P.M. VS. S.H.P. FOR EACH SHAFT
U.S.S. LEXINGTON & SARATOGA

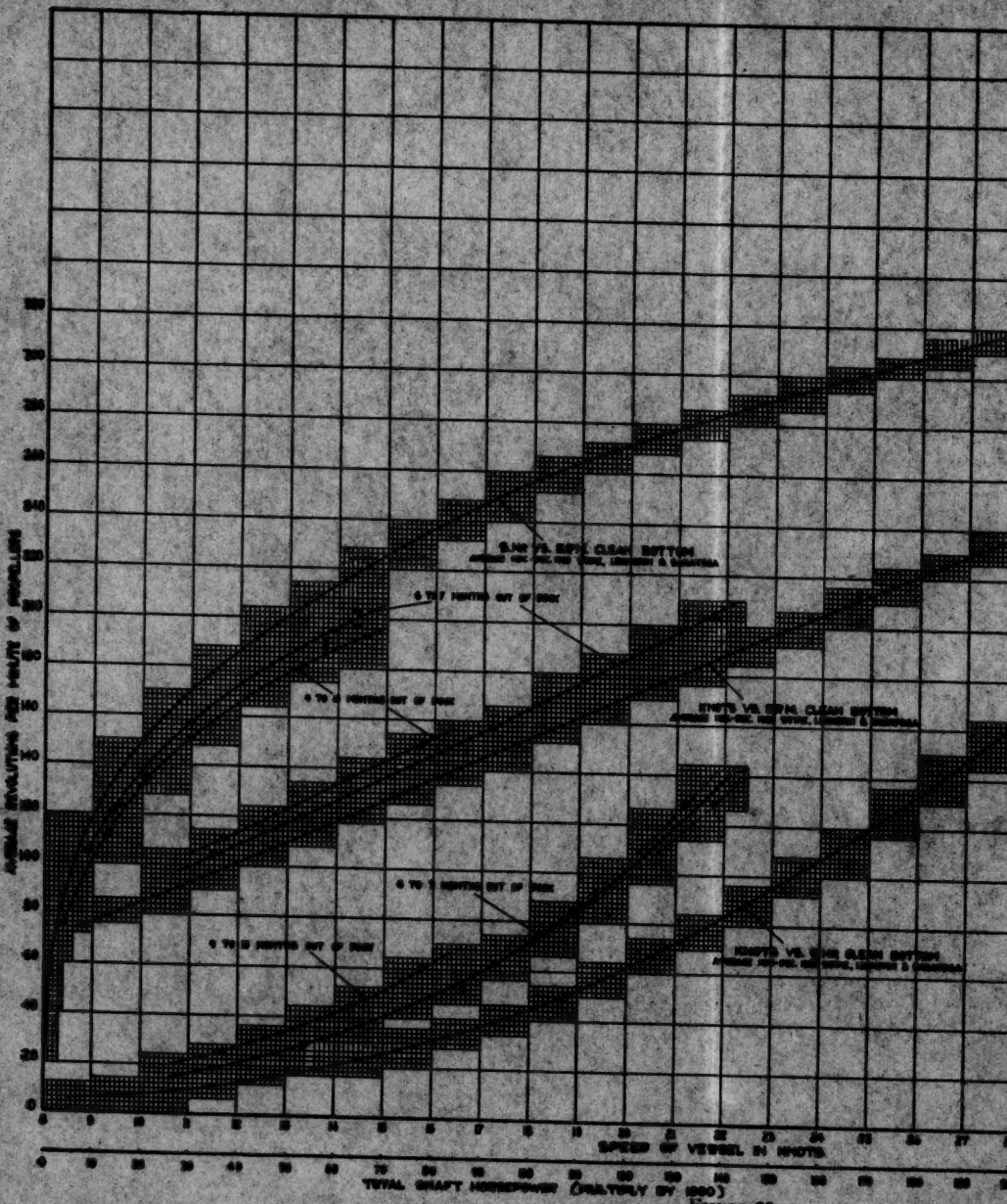
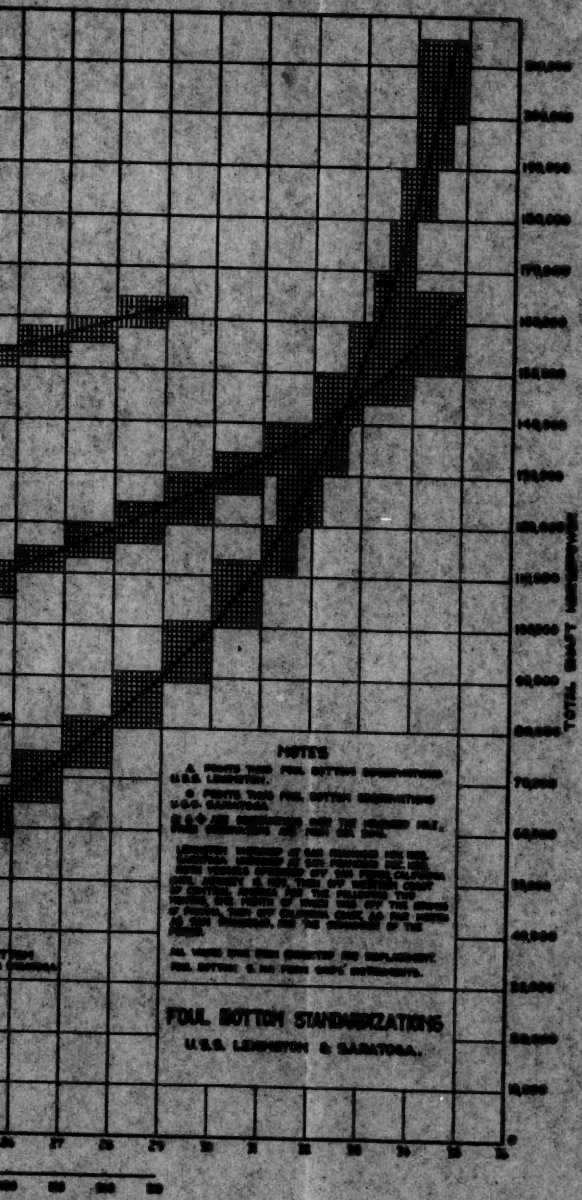
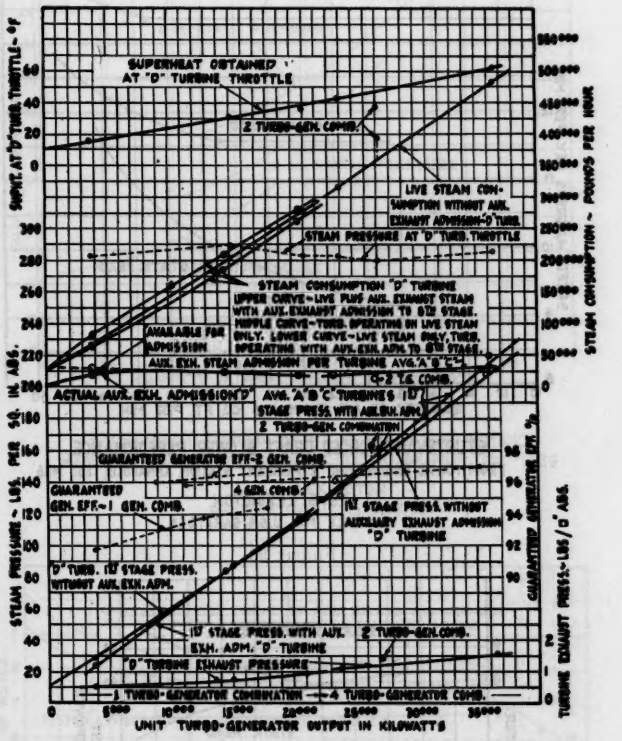

AVERAGE REVOLUTIONS PER MINUTE OF PROPELLERS
SPEED OF VESSEL IN KNOTS
TOTAL SHAFT HORSEPOWER (CORRECTED BY 1950)

FIGURE 22.


NOTES
FOUL BOTTOM STANDARDIZATIONS
U.S.S. LEXINGTON & SARATOGA.


SUPERHEAT OBTAINED AT "D" TURBINE THROTTLE
2 TURBO-GEN. COMBS.
LIVE STEAM CONSUMPTION WITHOUT AUX. EXHAUST ADMISSION-D' TURB.
STEAM PRESSURE AT "D" TURB. THROTTLE
STEAM CONSUMPTION "D" TURBINE
UPPER CURVE-LIVE PLUS AUX. EXHAUST STEAM WITH AUX. EXHAUST ADMISSION TO 8TH STAGE.
MIDDLE CURVE-TURB. OPERATING ON LIVE STEAM ONLY. LOWER CURVE-LIVE STEAM ONLY, TURB. OPERATING WITH AUX. EXH. ADM. TO 8TH STAGE.
AVAILABLE FOR ADMISSION
AUX. EXH. STEAM ADMISSION PER TURBINE AVG. "A" "B" "C"
C-2 T.G. COMBS.
ACTUAL AUX. EXH. ADMISSION "D"
AVG. "A" "B" "C" TURBINES 1ST STAGE PRESS. WITH AUX. EXH. ADM.
2 TURBO-GEN. COMBINATION
GUARANTEED GENERATOR EFF.-2 GEN. COMBS.
GUARANTEED GEN. EFF.-1 GEN. COMBS.
4 GEN. COMBS.
1ST STAGE PRESS. WITHOUT AUXILIARY EXHAUST ADMISSION "D" TURBINE
"D" TURB. 1ST STAGE PRESS. WITHOUT AUX. EXH. ADM.
1ST STAGE PRESS. WITH AUX. EXH. ADM. "D" TURBINE
2 TURBO-GEN. COMBS.
"D" TURBINE EXHAUST PRESSURE
1 TURBO-GENERATOR COMBINATION
4 TURBO-GENERATOR COMBS.
SUPHT. AT "D" TURB. THROTTLE - °F
STEAM PRESSURE - LBS. PER SQ. IN. ABS.
STEAM CONSUMPTION - POUNDS PER HOUR
GUARANTEED GENERATOR EFF. %
TURBINE EXHAUST PRESS. - LBS./□" ABS.
UNIT TURBO-GENERATOR OUTPUT IN KILOWATTS

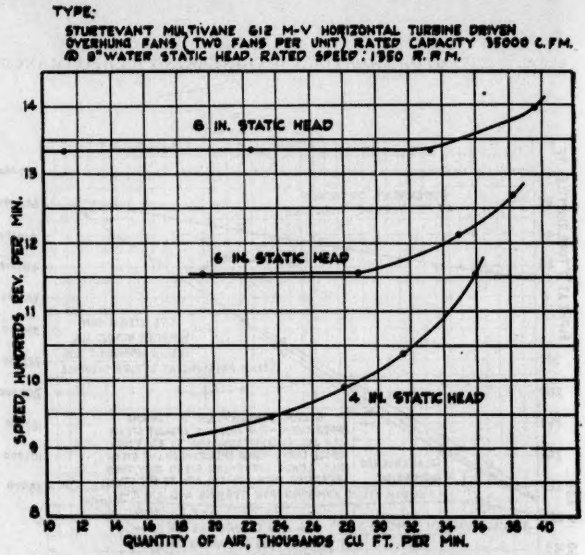
NOTE:-

ALL DATA ARE OBSERVED VALUES FROM OFFICIAL MACHINERY TRIALS OF NOV. 1928, EXCEPT AUXILIARY EXHAUST AND LIVE STEAM FLOW TO TURBINES WHILE OPERATING WITH AUXILIARY EXHAUST ADMISSION. THE EXHAUST ADMISSION FLOW TO TURBINES WAS CALCULATED FROM THE TURBINE ADMISSION BELT AND TURBINE STAGE PRESSURES OBTAINED DURING THE TRIALS USING A FLOW FACTOR OF 60% FOR THE EXHAUST ADMISSION ORIFICES. A CHECK OF THE EXHAUST ADMISSION TO TURBINES WAS MADE BY COMPUTING THE AUXILIARY EXHAUST AVAILABLE FOR TURBINE ADMISSION FROM THE EXHAUST STEAM DELIVERED TO THE AUXILIARY EXHAUST LINE (FROM TEST CURVES OF AUXILIARY MACHINERY) LESS AUXILIARY EXHAUST USED FOR FEED HEATING, GLAND SEALING AND DISTILLING PLANT. THE LIVE STEAM FLOW WITH TURBINE OPERATING WITH AUXILIARY EXHAUST ADMISSION WAS DETERMINED BY SUBTRACTING THE CALCULATED EXHAUST FLOW FROM THE METERED TOTAL STEAM FLOW THROUGH THE TURBINE.

POINTS THUS ⊙ FOR DESIGNATED TURBINES OPERATING UNDER OFFICIAL MACH. TRIAL CONDITIONS WITH AUX. EXH. ADM.
POINTS THUS ● FOR "D" TURBINE OPERATING UNDER OFFICIAL MACH. TRIAL CONDITIONS WITHOUT AUX. EXH. ADM.

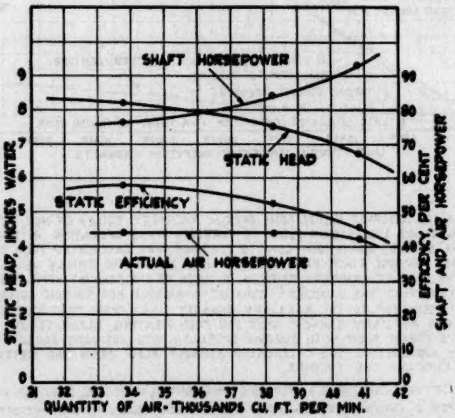
MAIN TURBINE PERFORMANCE
FROM OFFICIAL MACHINERY TRIALS OF NOV. 1928
U.S.S. LEXINGTON

FIGURE 18.


TYPE:
STURTEVANT MULTIVANE 612 M-V HORIZONTAL TURBINE DRIVEN OVERHUNG FANS (TWO FANS PER UNIT) RATED CAPACITY 38000 C.F.M. @ 8" WATER STATIC HEAD. RATED SPEED: 1350 R.P.M.
8 IN. STATIC HEAD
6 IN. STATIC HEAD
4 IN. STATIC HEAD
SPEED, HUNDREDS REV. PER MIN.
QUANTITY OF AIR, THOUSANDS CU. FT. PER MIN.

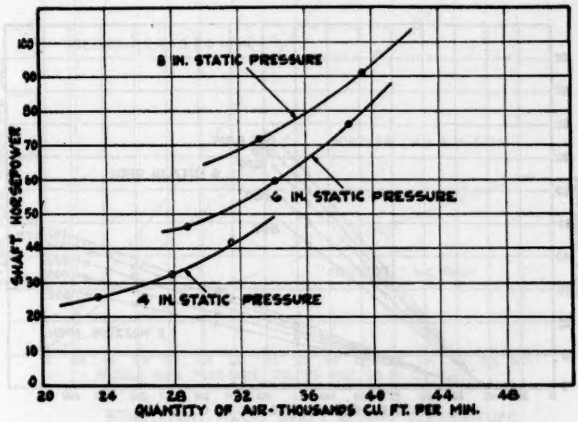
STURTEVANT FORCED DRAFT BLOWER PERFORMANCE
FROM NAVAL ENGINEERING EXPERIMENT STATION, ANNAPOLIS, TEST #1426
U.S.S. LEXINGTON
SUITABLE FOR U.S.S. SARATOGA

FIGURE 23.



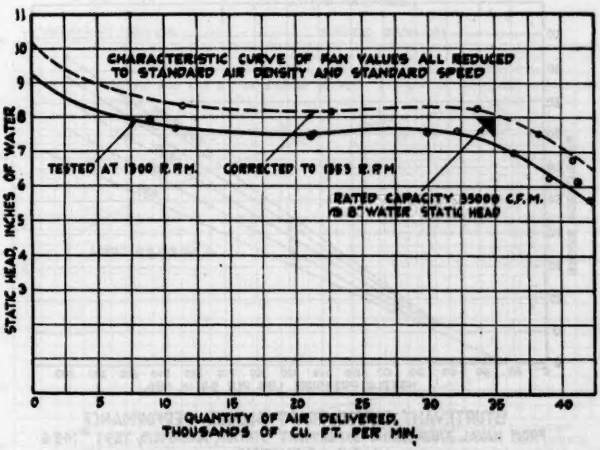
STURTEVANT FORCED DRAFT BLOWER PERFORMANCE
CONSTANT SPEED CHARACTERISTICS-ALL VALUES FOR 1350 R.P.M.
FROM NAVAL ENGINEERING EXPERIMENT STATION, ANNAPOLIS, TEST #1426
U.S.S. LEXINGTON
SUITABLE FOR U.S.S. SARATOGA

FIGURE 24.



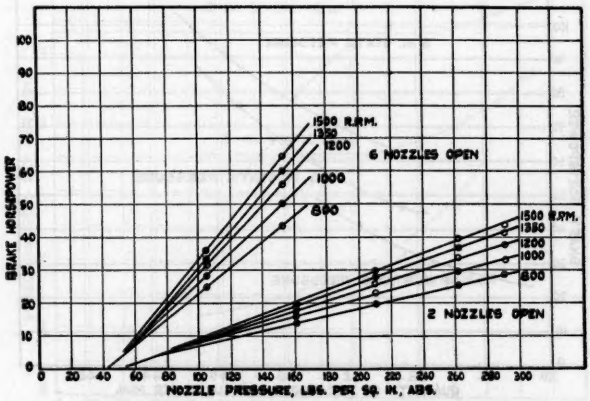
STURTEVANT FORCED DRAFT BLOWER PERFORMANCE
FROM NAVAL ENGINEERING EXPERIMENT STATION, ANNAPOLIS, TEST # 1426
U. S. S. LEXINGTON
SUITABLE FOR U.S.S. SARATOGA

FIGURE 25.


CHARACTERISTIC CURVE OF FAN VALUES ALL REDUCED TO STANDARD AIR DENSITY AND STANDARD SPEED
TESTED AT 1300 R.P.M.
CORRECTED TO 1363 R.P.M.
RATED CAPACITY 35000 C.F.M. @ 8" WATER STATIC HEAD
STATIC HEAD, INCHES OF WATER
QUANTITY OF AIR DELIVERED, THOUSANDS OF CU. FT. PER MIN.

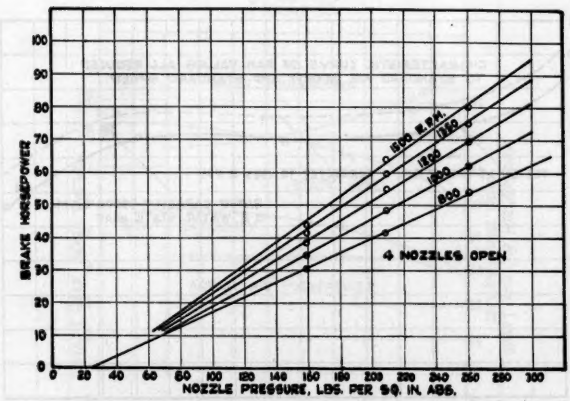
STURTEVANT FORCED DRAFT BLOWER PERFORMANCE
FROM NAVAL ENGINEERING EXPERIMENT STATION, ANNAPOLIS, TEST # 1426
U.S.S. LEXINGTON
SUITABLE FOR U.S.S. SARATOGA

FIGURE 26.



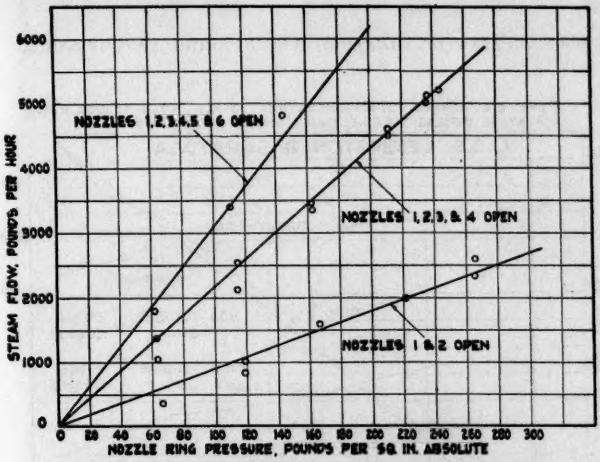
STURTEVANT FORCED DRAFT BLOWER PERFORMANCE
FROM NAVAL ENGINEERING EXPERIMENT STATION, ANNAPOLIS, TEST #1426
U.S.S. LEXINGTON
SUITABLE FOR U.S.S. SARATOGA

FIGURE 27.



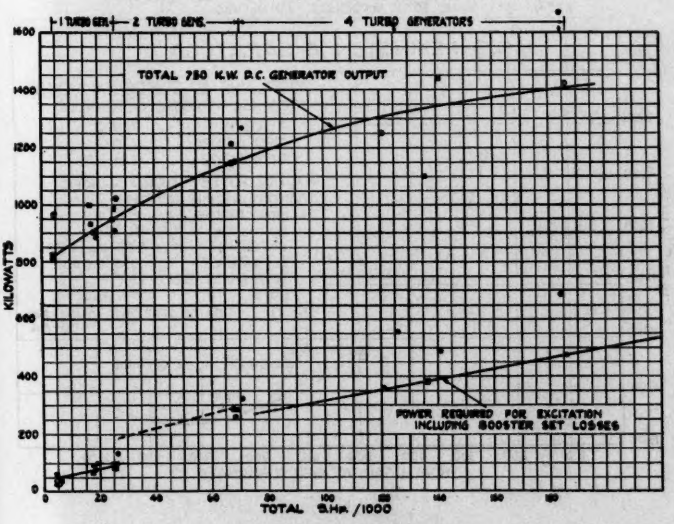
STURTEVANT FORCED DRAFT BLOWER PERFORMANCE
FROM NAVAL ENGINEERING EXPERIMENT STATION, ANNAPOLIS, TEST #1426
U.S.S. LEXINGTON
SUITABLE FOR U.S.S. SARATOGA

FIGURE 28.


STEAM FLOW, POUNDS PER HOUR
6000
5000
4000
3000
2000
1000
0
NOZZLES 1, 2, 3, 4, 5 & 6 OPEN
NOZZLES 1, 2, 3, & 4 OPEN
NOZZLES 1 & 2 OPEN
0 20 40 60 80 100 120 140 160 180 200 220 240 260 280 300
NOZZLE RING PRESSURE, POUNDS PER SQ. IN. ABSOLUTE

STURTEVANT FORCED DRAFT BLOWER PERFORMANCE
STEAM FLOW CURVES
FROM NAVAL ENGINEERING EXPERIMENT STATION, ANNAPOLIS, TEST #1426
U.S.S. LEXINGTON
SUITABLE FOR U.S.S. SARATOGA

FIGURE 29.



TOTAL D.C. POWER AND
D.C. POWER FOR EXCITATION.
U.S.S. LEXINGTON & SARATOGA

FIGURE 30.

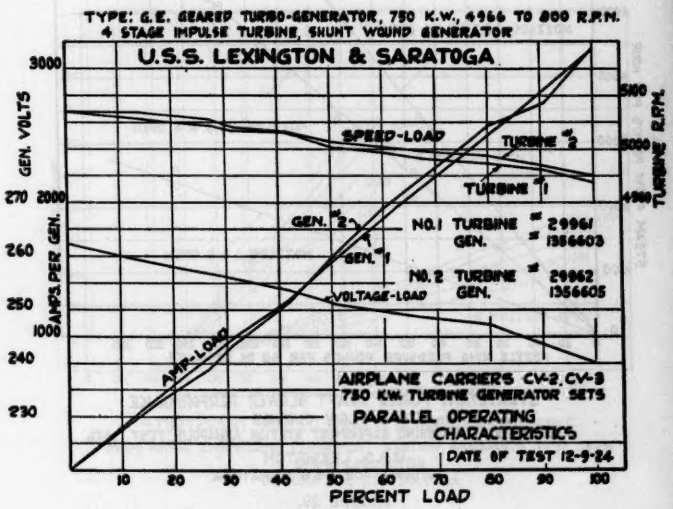


FIGURE 32.

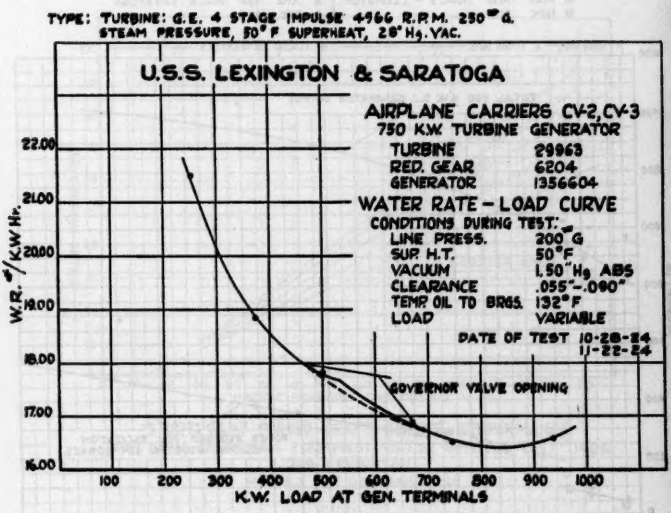


FIGURE 33.

TURBINE R.P.M.

43

5

E4
E4

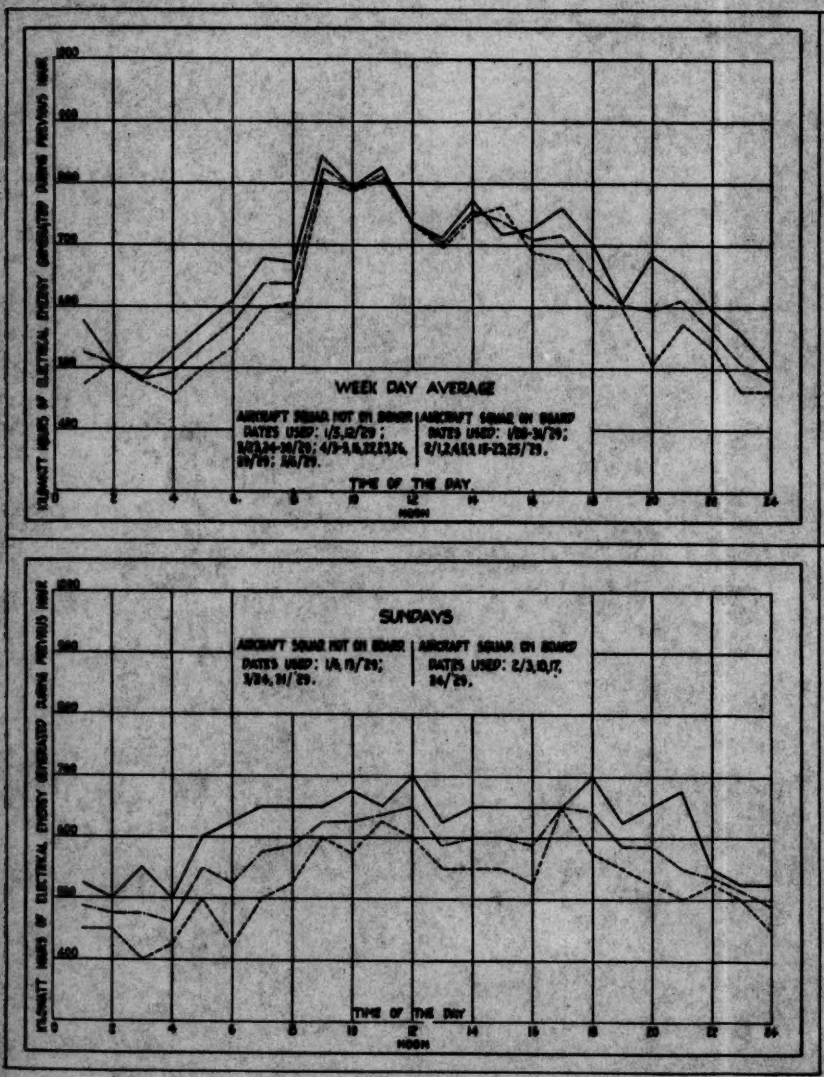
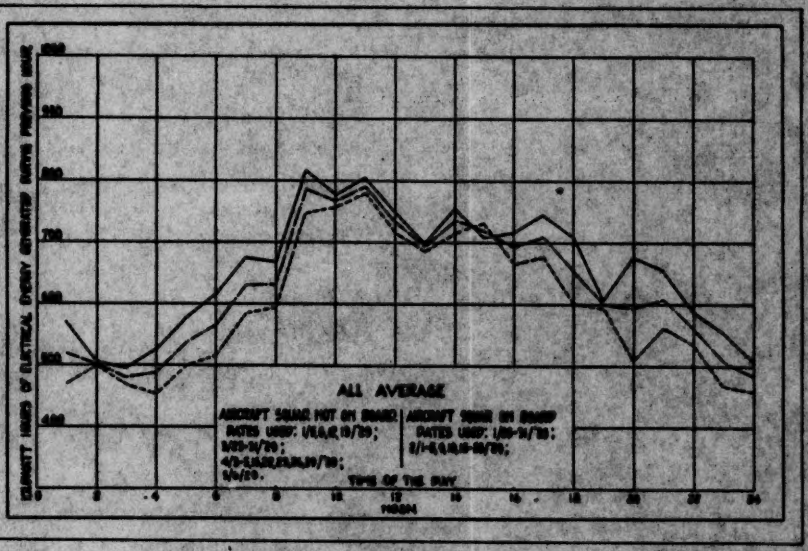

WEEK DAY AVERAGE
AIRCRAFT SQUAD NOT ON BOARD
DATES USED: 1/3,12/29;
AIRCRAFT SQUAD ON BOARD
DATES USED: 1/26-31/29;
TIME OF THE DAY
NOON
SUNDAYS
AIRCRAFT SQUAD NOT ON BOARD
DATES USED: 1/6,13/29;
AIRCRAFT SQUAD ON BOARD
DATES USED: 2/3,10,17,
24/29.
TIME OF THE DAY
NOON

FIGURE 31.



LEGEND

——— WITH AIRCRAFT SQUADRONS ON BOARD

- - - - - WITHOUT AIRCRAFT SQUADRONS ON BOARD

——— AVERAGE

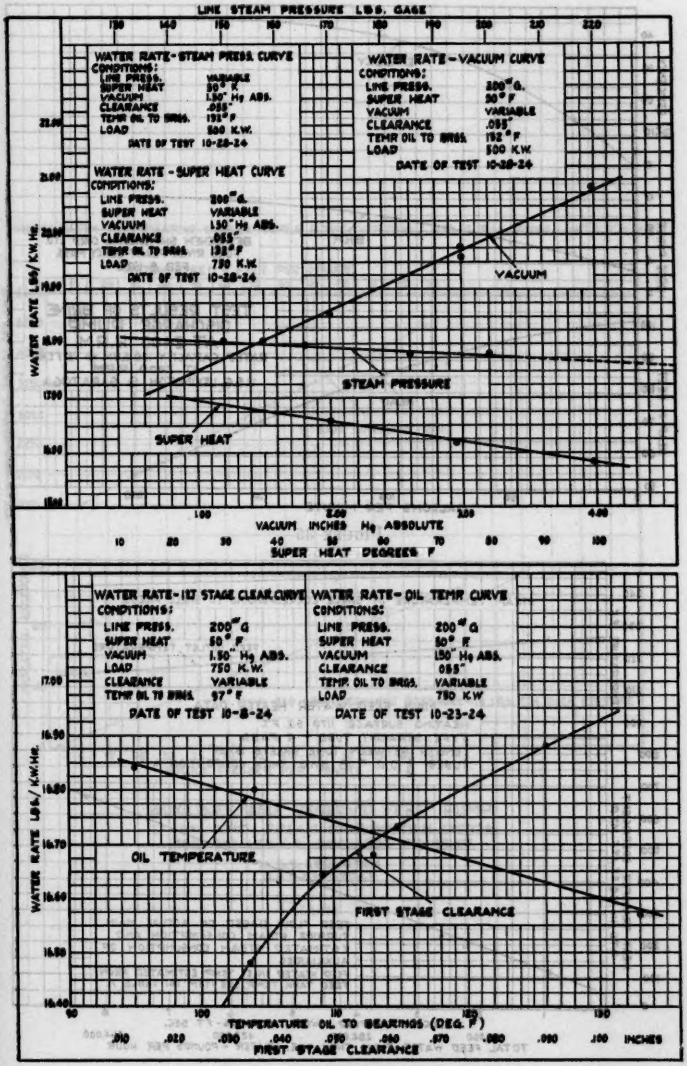
TOTAL D.C. LOAD CURVES

SHIP AT ANCHOR

U.S.S. LEXINGTON

FIGURE 31.

NOTE:- THESE CURVES ARE PLOTTED FROM TESTS ON TURBINE # 29963, REDUCTION GEAR # 6204, GENERATOR # 1356604 AT MANUFACTURERS WORKS



WATER RATE CORRECTION CURVES
FOR
750 K.W. TURBO-GENERATORS
U.S.S. LEXINGTON & SARATOGA

FIGURE 34.

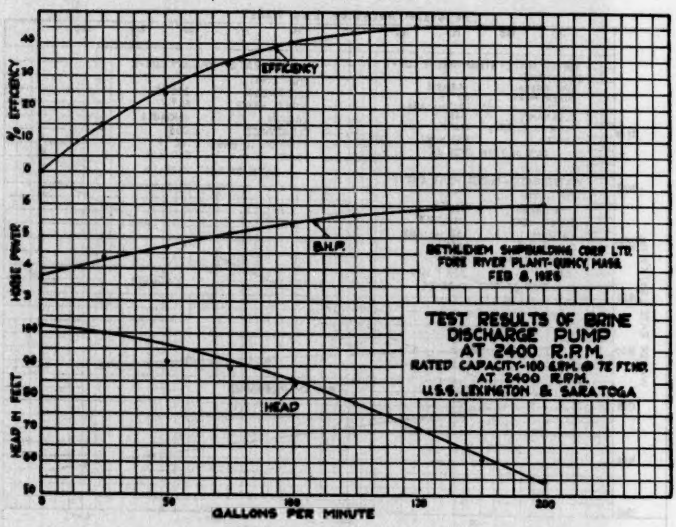
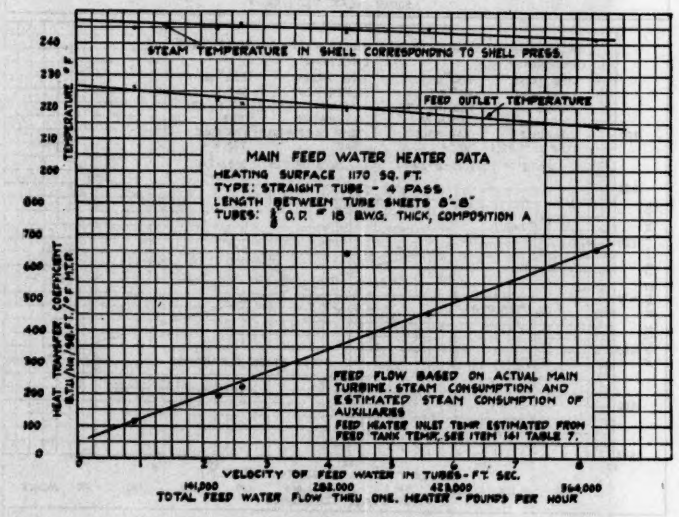
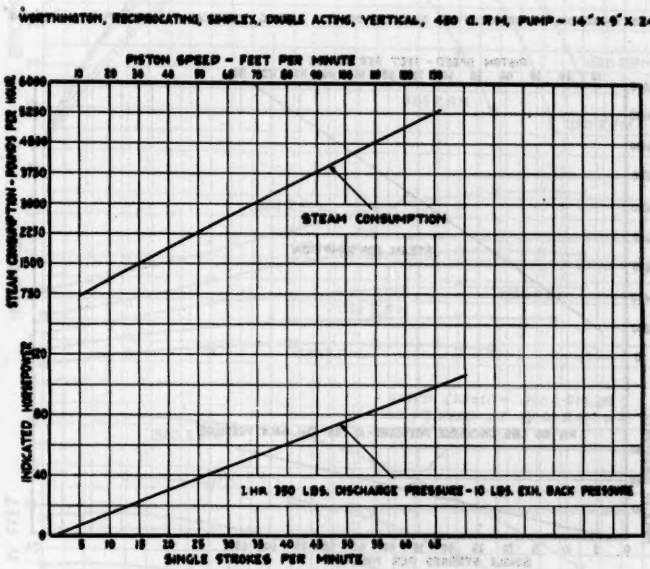


FIGURE 35


STEAM TEMPERATURE IN SHELL CORRESPONDING TO SHELL PRESS.
FEED OUTLET TEMPERATURE
TEMPERATURE °F
MAIN FEED WATER HEATER DATA
HEATING SURFACE 1170 SQ. FT.
TYPE: STRAIGHT TUBE - 4 PASS
LENGTH BETWEEN TUBE SHEETS 8'-8"
TUBES: ⅝ O.D. " 18 B.W.G. THICK, COMPOSITION A
HEAT TRANSFER COEFFICIENT B.T.U./hr./sq.ft./°F DIFF.
FEED FLOW BASED ON ACTUAL MAIN TURBINE STEAM CONSUMPTION AND ESTIMATED STEAM CONSUMPTION OF AUXILIARIES
FEED HEATER INLET TEMP ESTIMATED FROM FEED TANK TEMP. SEE ITEM 141 TABLE 7.
VELOCITY OF FEED WATER IN TUBES - FT. SEC.
141,000
288,000
423,000
564,000
TOTAL FEED WATER FLOW THRU ONE HEATER - POUNDS PER HOUR

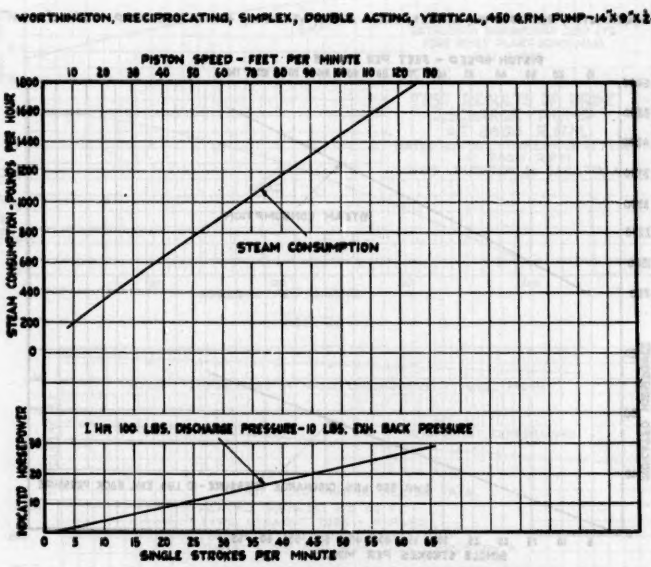
MAIN FEED WATER HEATER
SERVICE PERFORMANCE
FROM
NOV. 1928 ACCEPTANCE TRIAL DATA
U.S.S. LEXINGTON.

FIGURE 36.


WORTHINGTON, RECIPROCATING, SIMPLEX, DOUBLE ACTING, VERTICAL, 480 G. P. M. PUMP - 14" X 9" X 24"
PISTON SPEED - FEET PER MINUTE
STEAM CONSUMPTION - POUNDS PER HOUR
INDICATED HORSEPOWER
STEAM CONSUMPTION
I.HR. 350 LBS. DISCHARGE PRESSURE - 10 LBS. EXH. BACK PRESSURE
SINGLE STROKES PER MINUTE

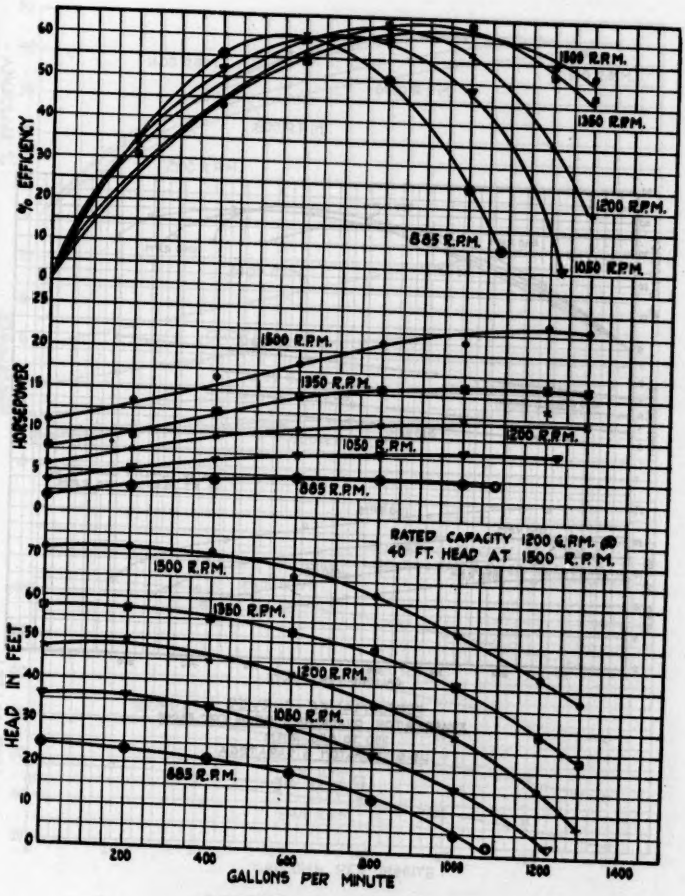
AUXILIARY AND RESERVE FEED PUMP
I. HP. AND STEAM CONSUMPTION CURVES FOR 350 LBS. / □" DISCHARGE PRESSURE
U. S. S. LEXINGTON
U. S. S. SARATOGA

FIGURE 37.


WORTHINGTON, RECIPROCATING, SIMPLEX, DOUBLE ACTING, VERTICAL, 450 G.P.M. PUMP - 14"x9"x24"
PISTON SPEED - FEET PER MINUTE
10 20 30 40 50 60 70 80 90 100 110 120 130
STEAM CONSUMPTION - POUNDS PER HOUR
1800 1600 1400 1200 1000 800 600 400 200 0
STEAM CONSUMPTION
I. H.P.
INDICATED HORSEPOWER
30 20 10 0
I. HP. 100 LBS. DISCHARGE PRESSURE - 10 LBS. EXH. BACK PRESSURE
0 5 10 15 20 25 30 35 40 45 50 55 60 65
SINGLE STROKES PER MINUTE

AUXILIARY AND RESERVE FEED PUMP
I. HP. AND STEAM CONSUMPTION CURVES
FOR 100 LBS. / D" DISCHARGE PRESSURE
U.S.S. LEXINGTON
U.S.S. SARATOGA

FIGURE 38.



PERFORMANCE CURVES
DISTILLER CIRCULATING PUMP
1500 TO 885 R.P.M.
U.S.S. LEXINGTON & SARATOGA

FIGURE 39.

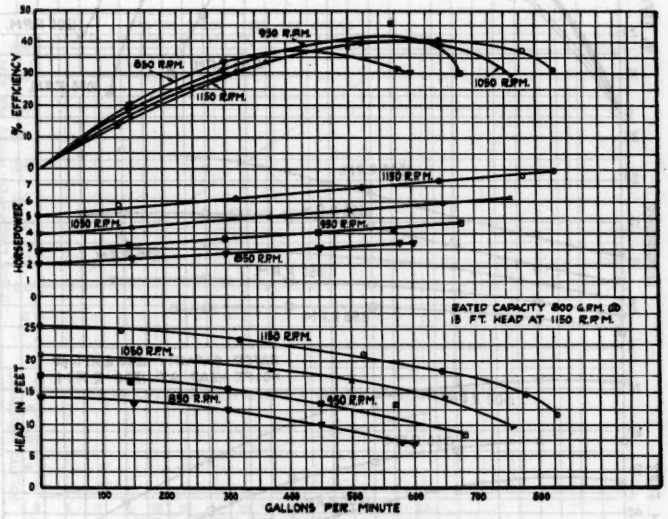
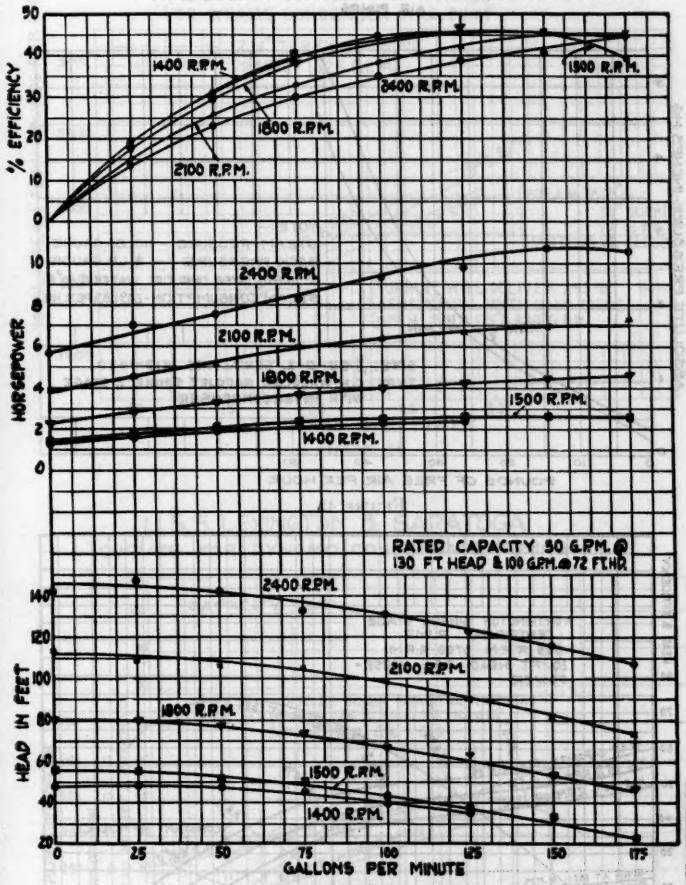

% EFFICIENCY
930 R.P.M.
850 R.P.M.
1150 R.P.M.
1050 R.P.M.
HORSEPOWER
1150 R.P.M.
1050 R.P.M.
930 R.P.M.
850 R.P.M.
RATED CAPACITY 800 G.P.M. @ 15 FT. HEAD AT 1150 R.P.M.
HEAD IN FEET
1150 R.P.M.
1050 R.P.M.
850 R.P.M.
930 R.P.M.
GALLONS PER MINUTE
PERFORMANCE CURVES
EVAPORATOR QUADRUPLE BRINE CIRC. PUMP
1150 TO 850 R.P.M
U.S.S. LEXINGTON & SARATOGA

FIGURE 40.



PERFORMANCE CURVES
DISTILLER FRESH WATER PUMP
2400 TO 1400 R.P.M.
U.S.S. LEXINGTON & SARATOGA

FIGURE 41.

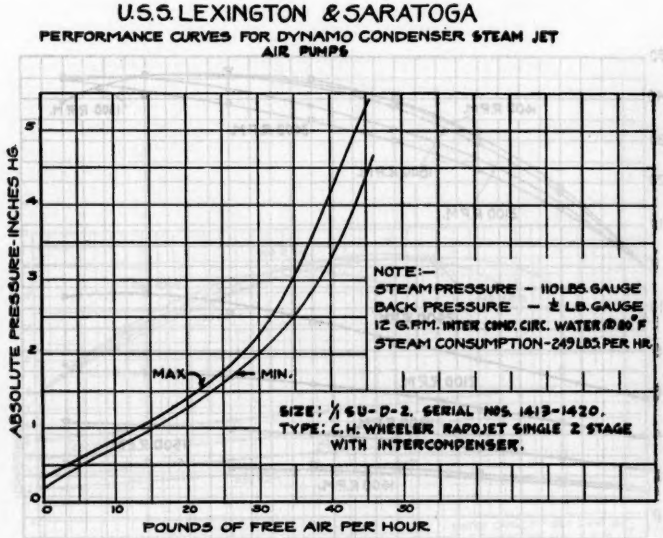
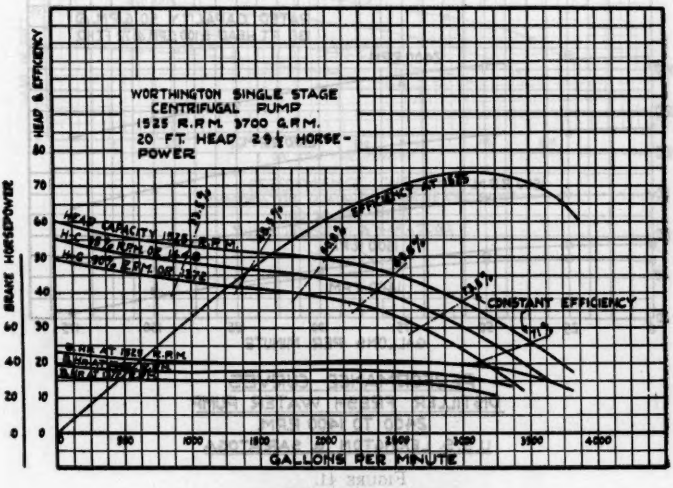

U.S.S. LEXINGTON & SARATOGA
PERFORMANCE CURVES FOR DYNAMO CONDENSER STEAM JET AIR PUMPS
ABSOLUTE PRESSURE-INCHES HG.
POUNDS OF FREE AIR PER HOUR
MAX.
MIN.
NOTE:—
STEAM PRESSURE – 110 LBS. GAUGE
BACK PRESSURE – ½ LB. GAUGE
12 G.P.M. INTER COND. CIRC. WATER @ 80° F
STEAM CONSUMPTION – 249 LBS. PER HR.
SIZE: 7½ GU-D-2. SERIAL NOS. 1413-1420.
TYPE: C.H. WHEELER RAPOJET SINGLE 2 STAGE WITH INTERCONDENSER.

FIGURE 42.



PERFORMANCE CURVES
DYNAMO CONDENSER CIRCULATING PUMP
U.S.S. LEXINGTON & SARATOGA

FIGURE 43.

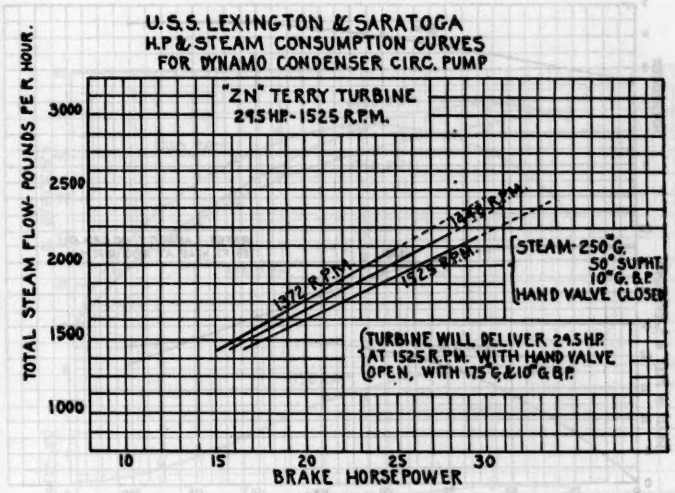


FIGURE 44.

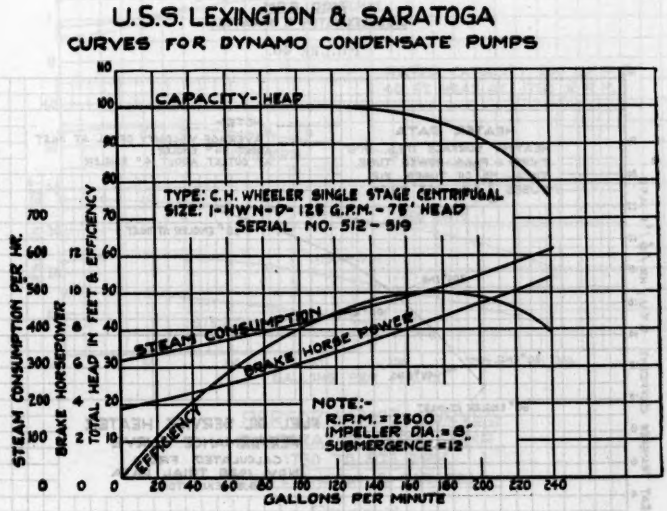


FIGURE 45.

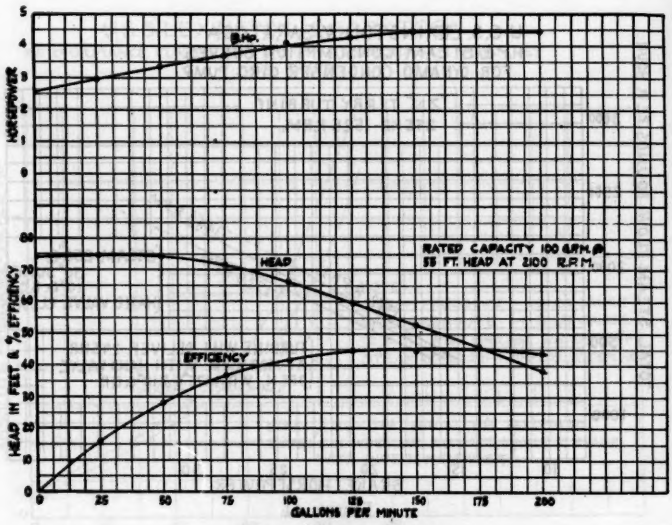


FIGURE 46.

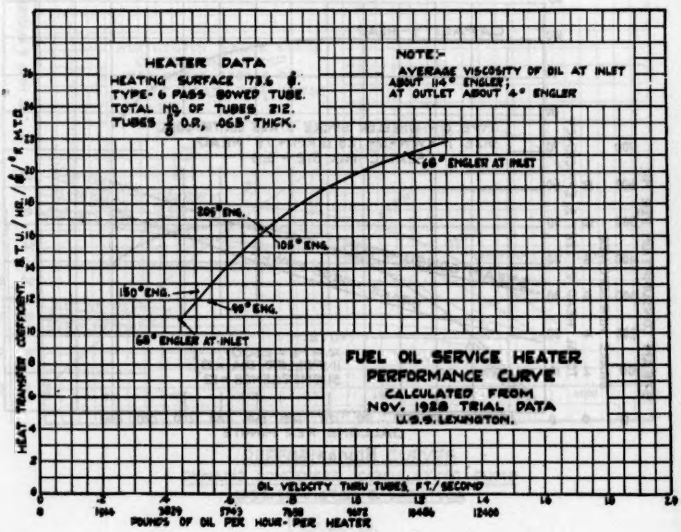
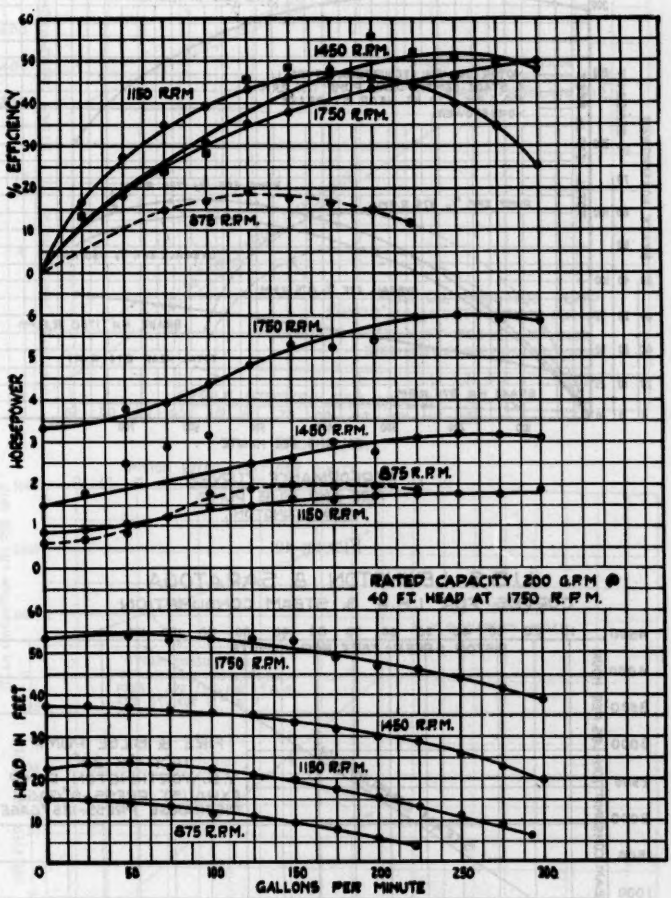
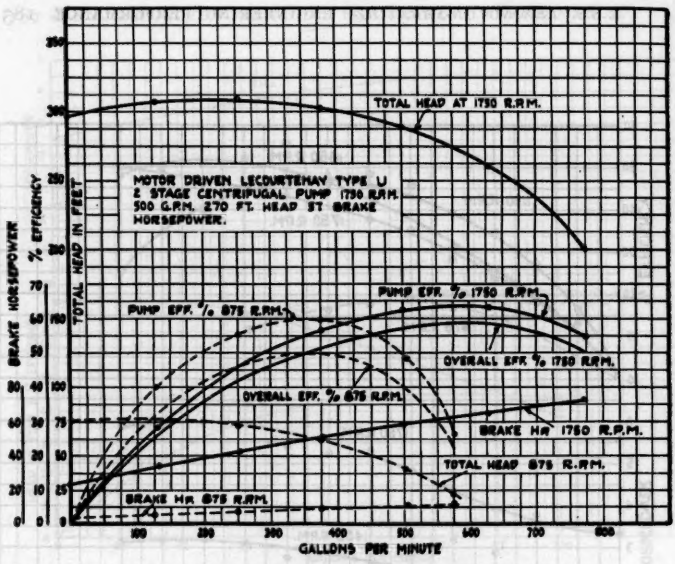


FIGURE 47.



PERFORMANCE CURVES
EVAPORATOR FEED PUMP
1750 TO 875 R.P.M.
U.S.S. LEXINGTON & SARATOGA.

FIGURE 48.


TOTAL HEAD AT 1750 R.P.M.
MOTOR DRIVEN LECOURTENAY TYPE U
2 STAGE CENTRIFUGAL PUMP 1750 R.P.M.
500 G.P.M. 270 FT. HEAD 37 BRAKE
HORSEPOWER.
PUMP EFF. % 1750 R.P.M.
PUMP EFF. % 875 R.P.M.
OVERALL EFF. % 1750 R.P.M.
OVERALL EFF. % 875 R.P.M.
BRAKE HP 1750 R.P.M.
TOTAL HEAD 875 R.P.M.
BRAKE HP 875 R.P.M.
BRAKE HORSEPOWER
% EFFICIENCY
TOTAL HEAD IN FEET
GALLONS PER MINUTE

PERFORMANCE CURVES
FIRE AND BILGE PUMP
U.S.S. LEXINGTON.

FIGURE 49.

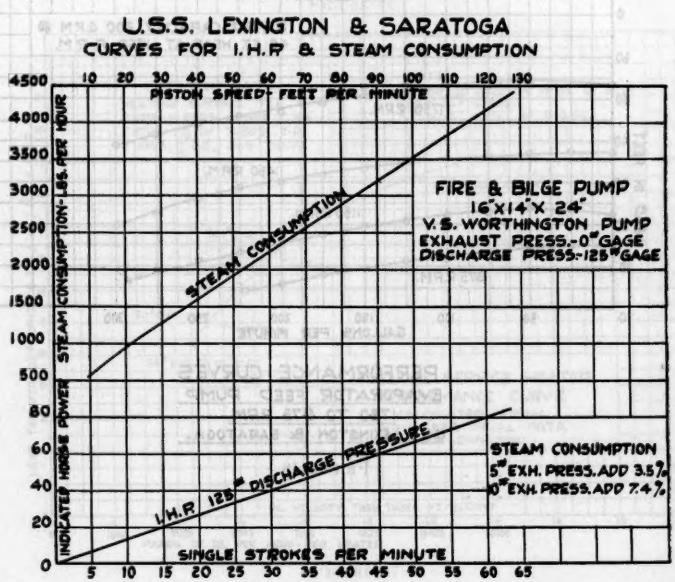


FIGURE 50.

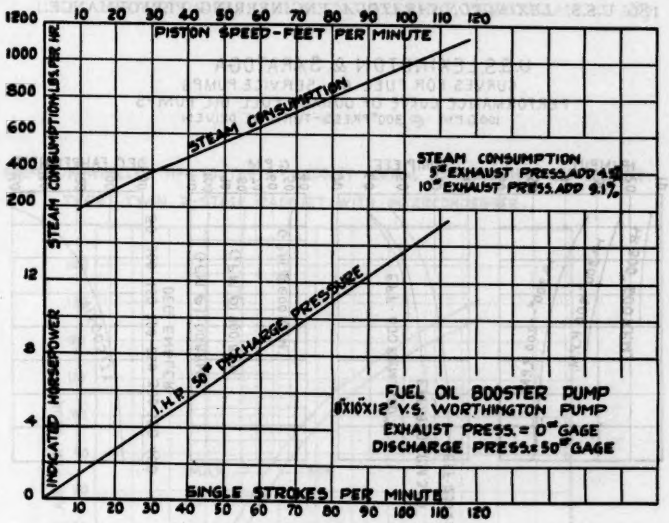
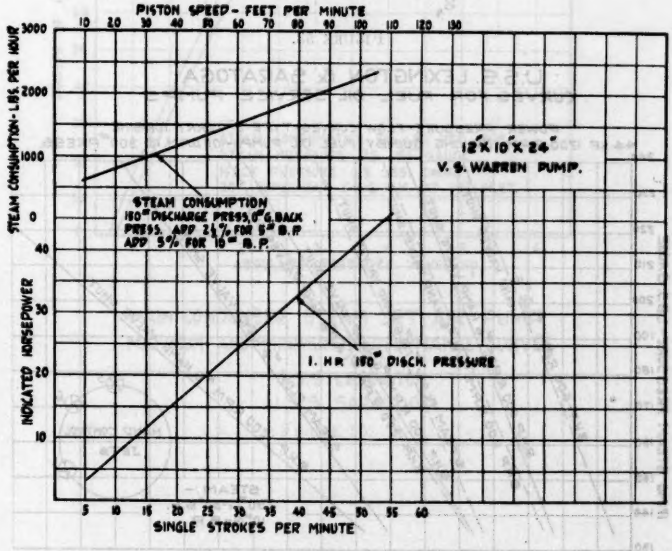

U.S.S. LEXINGTON & SARATOGA
CURVES FOR I.H.P & STEAM CONSUMPTION
PISTON SPEED - FEET PER MINUTE
STEAM CONSUMPTION LBS. PER HR.
INDICATED HORSEPOWER
STEAM CONSUMPTION
STEAM CONSUMPTION
5# EXHAUST PRESS. ADD 4%
10# EXHAUST PRESS. ADD 9.1%
I.H.P. 50# DISCHARGE PRESSURE
FUEL OIL BOOSTER PUMP
6X10X12" V.S. WORTHINGTON PUMP
EXHAUST PRESS. = 0# GAGE
DISCHARGE PRESS. = 50# GAGE
SINGLE STROKES PER MINUTE

FIGURE 51.


PISTON SPEED - FEET PER MINUTE
STEAM CONSUMPTION - LBS. PER HOUR
INDICATED HORSEPOWER
12"X 10"X 24"
V. S. WARREN PUMP.
STEAM CONSUMPTION
150# DISCHARGE PRESS. 0# G. BACK PRESS. ADD 2 1/2% FOR 5# B.P
ADD 5% FOR 10# B.P.
I. HR 150# DISCH. PRESSURE
SINGLE STROKES PER MINUTE

FUEL OIL RECIRCULATING PUMP
I. HR AND STEAM CONSUMPTION CURVES
U.S.S. LEXINGTON & SARATOGA

FIGURE 52.

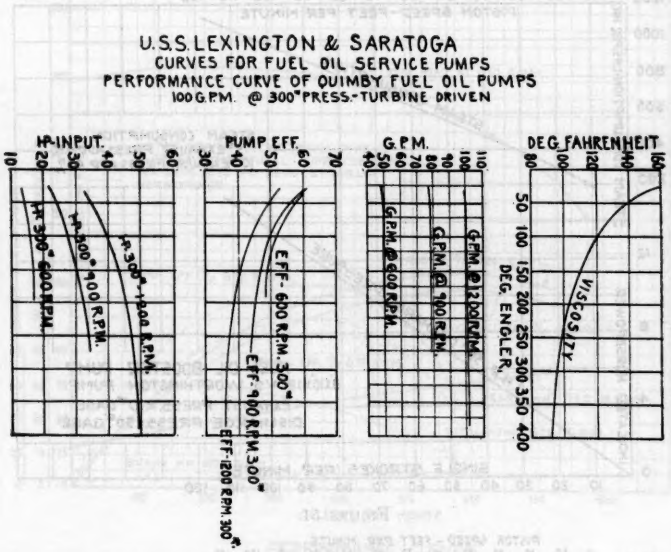


FIGURE 53.

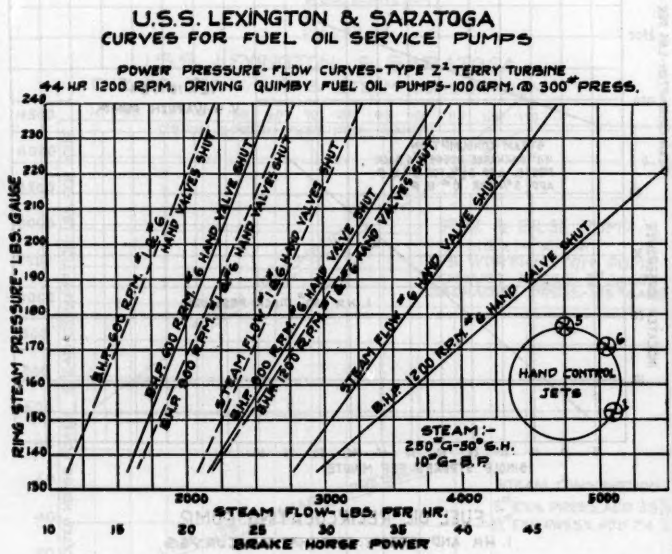
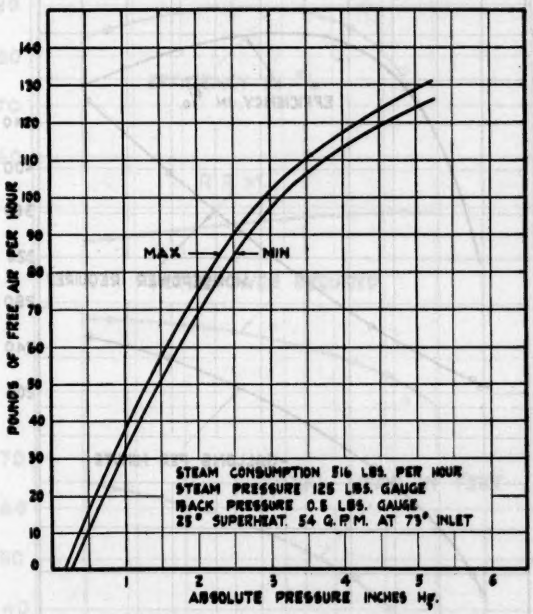

U.S.S. LEXINGTON & SARATOGA
CURVES FOR FUEL OIL SERVICE PUMPS
POWER PRESSURE-FLOW CURVES-TYPE Z¹ TERRY TURBINE
44 H.P. 1200 R.P.M. DRIVING QUIMBY FUEL OIL PUMPS-100 G.P.M. @ 300# PRESS.
RING STEAM PRESSURE-LBS. GAUGE
STEAM FLOW-LBS. PER HR.
BRAKE HORSE POWER
B.H.P. 600 R.P.M. "G" HAND VALVES SHUT
B.H.P. 900 R.P.M. "G" HAND VALVE SHUT
B.H.P. 1200 R.P.M. "G" HAND VALVE SHUT
STEAM FLOW "G" HAND VALVE SHUT
HAND CONTROL JETS
STEAM:- 250°F-50° S.H. 10# G-B.P.

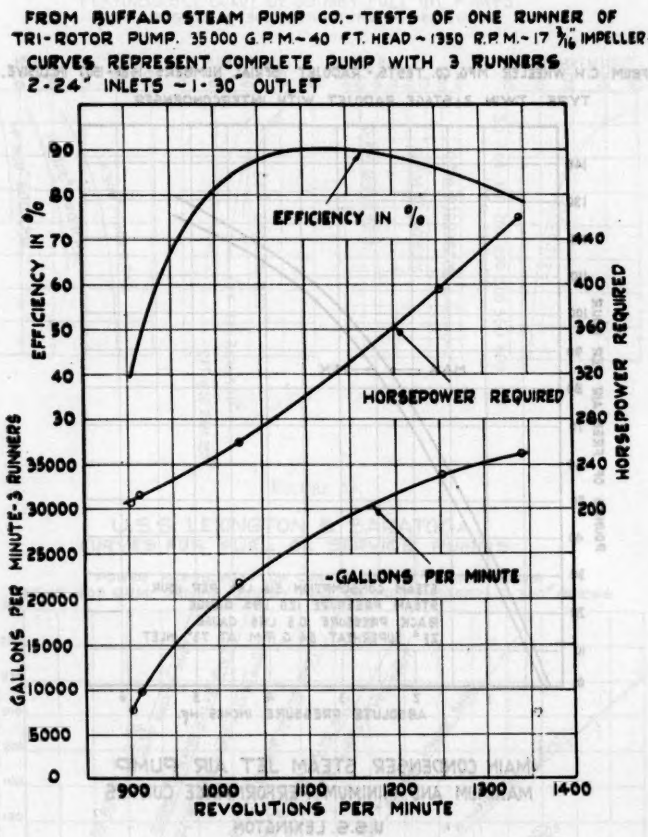
FIGURE 54.

FROM C.H. WHEELER MFG. CO. TESTS. · RADOJET SERIAL NUMBERS 1488-1507 INCLUSIVE.
TYPE: TWIN 2-STAGE RADOJET WITH INTERCONDENSER.



MAIN CONDENSER STEAM JET AIR PUMP
MAXIMUM AND MINIMUM PERFORMANCE CURVES
U.S.S. LEXINGTON
U.S.S. SARATOGA

FIGURE 55.



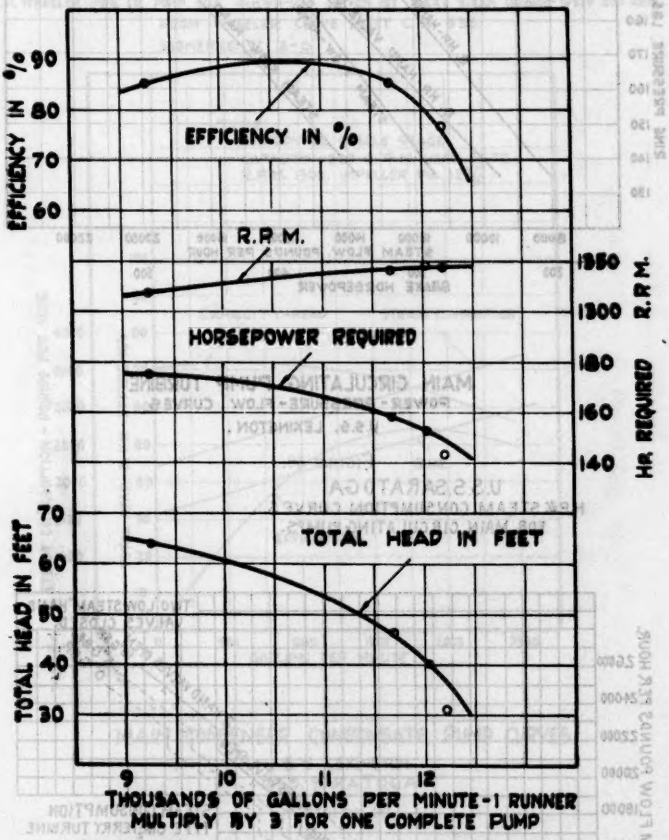
MAIN CIRCULATING PUMP CHARACTERISTICS
CONSTANT TOTAL HEAD 40 FT. - VARIABLE SPEED

U.S.S. LEXINGTON

FIGURE 56.

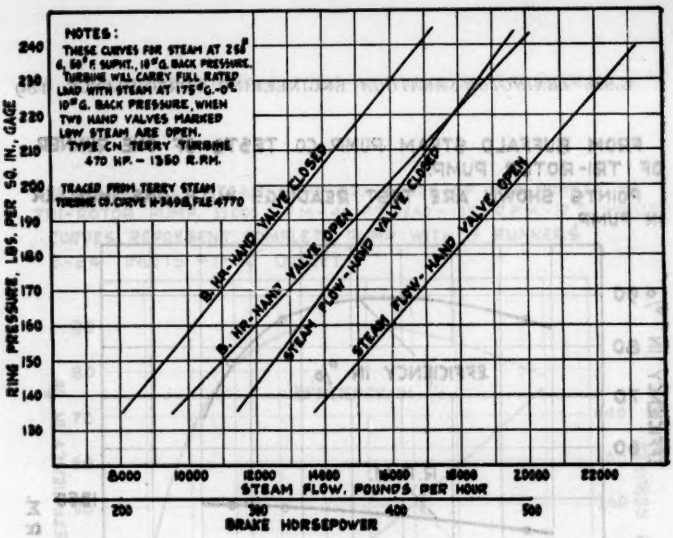
FROM BUFFALO STEAM PUMP CO., TESTS OF ONE RUNNER OF TRI-ROTOR PUMP.

POINTS SHOWN ARE TEST READINGS WITH ONE IMPELLER IN PUMP.



MAIN CIRCULATING PUMP CHARACTERISTICS
VARIABLE DISCHARGE HEAD-SUCTION HEAD 25 FEET
U.S.S. LEXINGTON

FIGURE 57.



MAIN CIRCULATING PUMP TURBINE
POWER - PRESSURE - FLOW CURVES
U.S.S. LEXINGTON.

FIGURE 58.

U.S.S. SARATOGA
H.P. & STEAM CONSUMPTION CURVES
FOR MAIN CIRCULATING PUMPS.

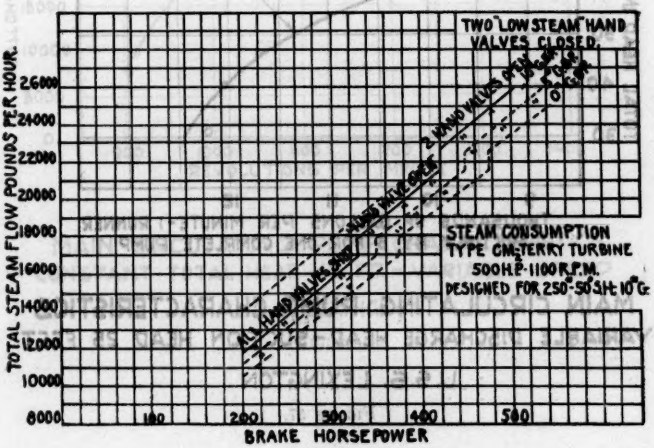
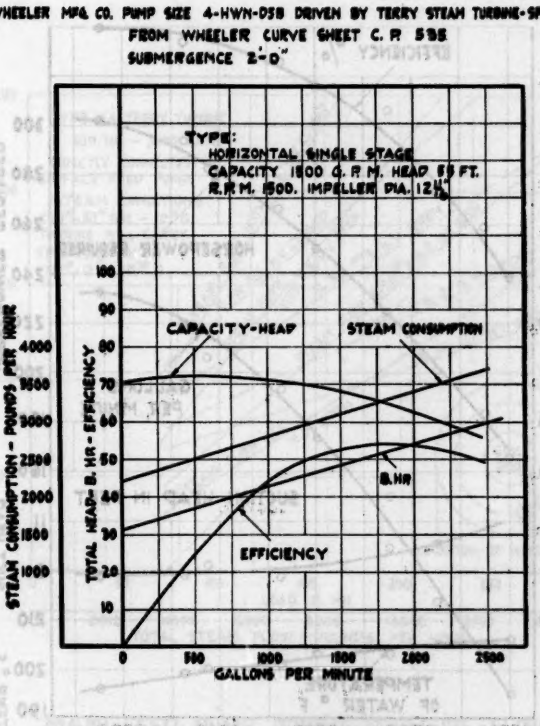


FIGURE 59.



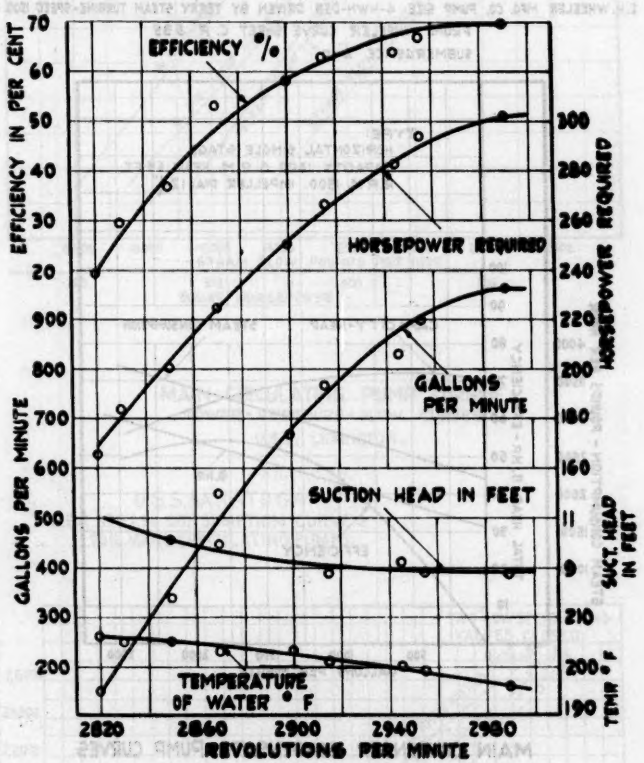
MAIN CONDENSER CONDENSATE PUMP CURVES

U.S.S. LEXINGTON
U.S.S. SARATOGA

FIGURE 60.

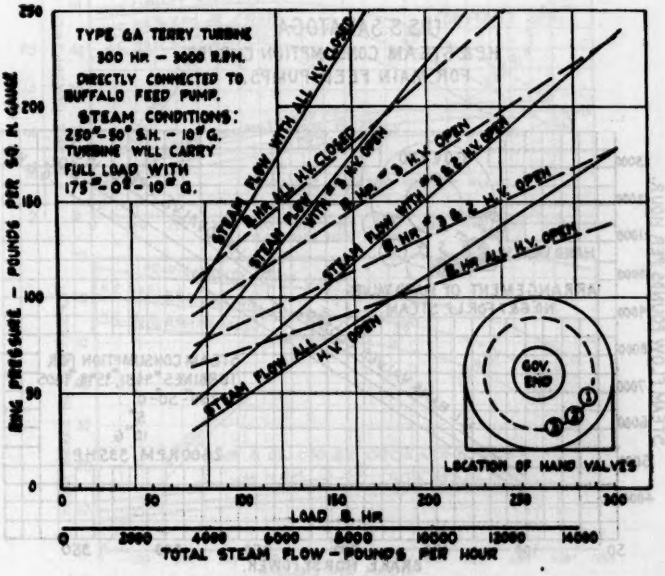
PUMP DATA;

BUFFALO CENTRIFUGAL FEED ~ 900 G.P.M., 900 FT. HEAD., 3000 R.P.M. ~ 4 STAGE ~ 8" INLET ~ 6" OUTLET ~ 12" AND 11½" IMPELLER DIAMETERS.



MAIN FEED PUMP CHARACTERISTICS
FROM BUFFALO STEAM PUMP CO. TESTS
CONSTANT DISCHARGE HEAD OF 900 FEET-VARIABLE SPEED
U.S.S. LEXINGTON

FIGURE 61.


TYPE GA TERRY TURBINE
300 HR – 3000 R.P.M.
DIRECTLY CONNECTED TO BUFFALO FEED PUMP.
STEAM CONDITIONS:
250°–50° S.H. – 10" G.
TURBINE WILL CARRY FULL LOAD WITH
175°–0° – 10" G.
STEAM FLOW WITH ALL H.V. CLOSED
STEAM FLOW WITH 1 H.V. OPEN
STEAM FLOW WITH 2 H.V. OPEN
STEAM FLOW WITH 3 H.V. OPEN
STEAM FLOW ALL H.V. OPEN
GOV. END
LOCATION OF HAND VALVES
RING PRESSURE – POUNDS PER SQ. IN. GAUGE
LOAD S. HP.
TOTAL STEAM FLOW – POUNDS PER HOUR

MAIN FEED PUMP POWER-PRESSURE-FLOW CURVES

U. S. S. LEXINGTON

FIGURE 62.

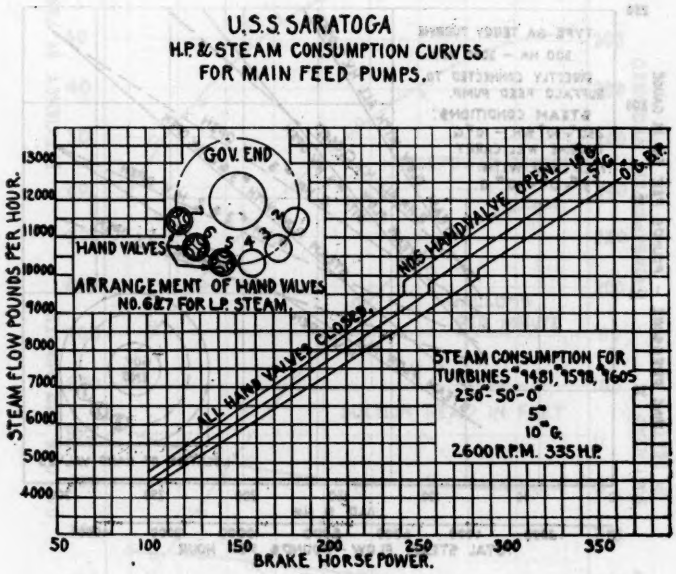
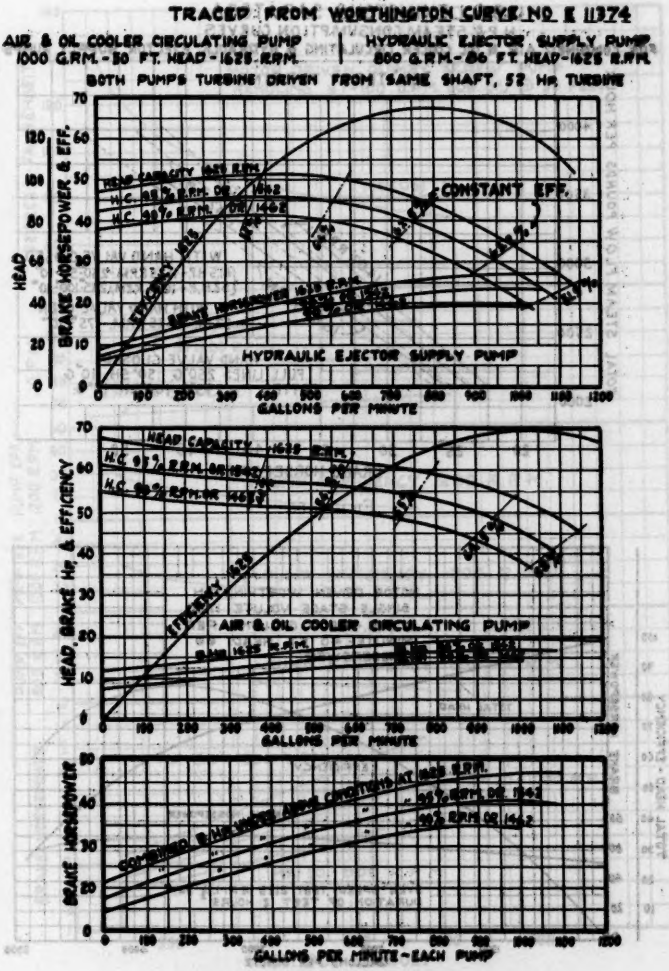

U.S.S. SARATOGA
H.P. & STEAM CONSUMPTION CURVES
FOR MAIN FEED PUMPS.
GOV. END
HAND VALVES
ARRANGEMENT OF HAND VALVES
NO. 6&7 FOR L.P. STEAM.
ALL HAND VALVES CLOSED.
NOS. 1 HAND VALVE OPEN
STEAM CONSUMPTION FOR
TURBINES 1481, 1598, 1605
250 - 50 - 0
5
10 G.
2600 R.P.M. 335 H.P.
STEAM FLOW POUNDS PER HOUR.
13000
12000
11000
10000
9000
8000
7000
6000
5000
4000
50
100
150
200
250
300
350
BRAKE HORSE POWER.

FIGURE 63.


TRACED FROM WORTHINGTON CURVE NO. E 11374
AIR & OIL COOLER CIRCULATING PUMP
1000 G.P.M. - 30 FT. HEAD - 1625 R.P.M.
HYDRAULIC EJECTOR SUPPLY PUMP
800 G.P.M. - 86 FT. HEAD - 1625 R.P.M.
BOTH PUMPS TURBINE DRIVEN FROM SAME SHAFT, 52 HP. TURBINE
HEAD
BRAKE HORSEPOWER & EFF.
CONSTANT EFF.
HYDRAULIC EJECTOR SUPPLY PUMP
GALLONS PER MINUTE
HEAD, BRAKE HP. & EFFICIENCY
AIR & OIL COOLER CIRCULATING PUMP
GALLONS PER MINUTE
BRAKE HORSEPOWER
GALLONS PER MINUTE - EACH PUMP

PERFORMANCE CURVES
AIR & OIL COOLER CIRCULATING & HYDRAULIC
EJECTOR SUPPLY PUMPS
U. S. S. LEXINGTON,
U. S. S. SARATOGA.

FIGURE 64.

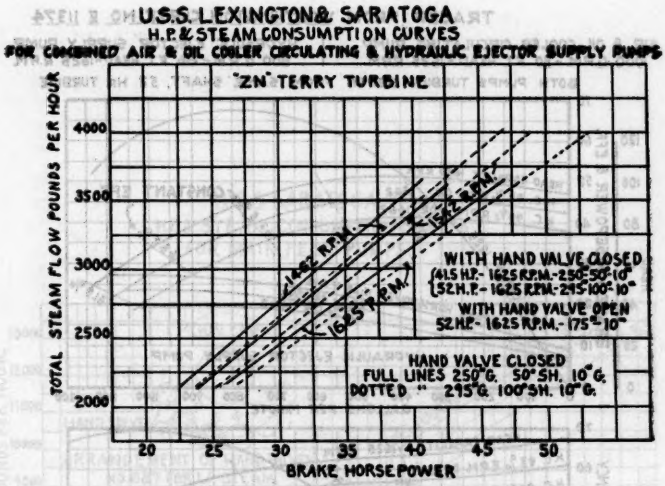
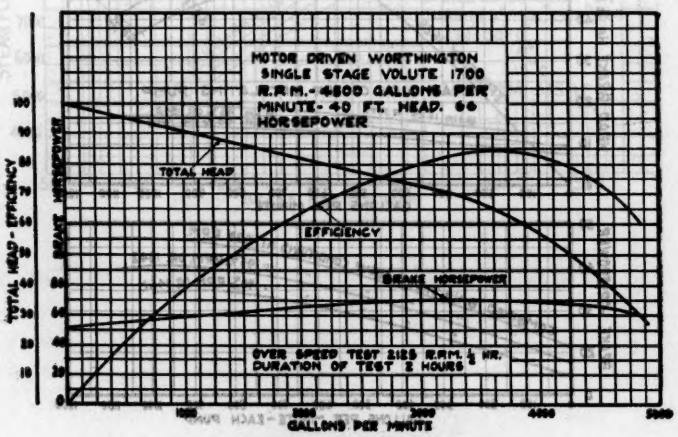
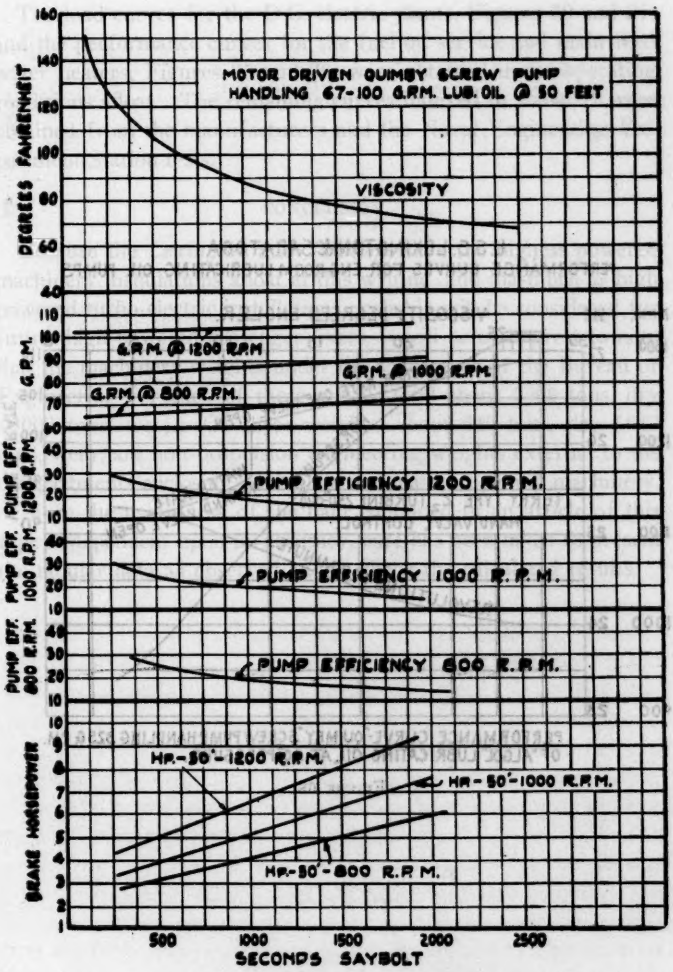


FIGURE 65.



PERFORMANCE CURVES
AIR & OIL COOLER CIRCULATING PUMP
(IN MOTOR ROOM)
U.S.S. LEXINGTON

FIGURE 66.



PERFORMANCE CURVES
MAIN MOTOR ROOM
LUBRICATING OIL PUMPS
U.S.S. LEXINGTON.

FIGURE 67.

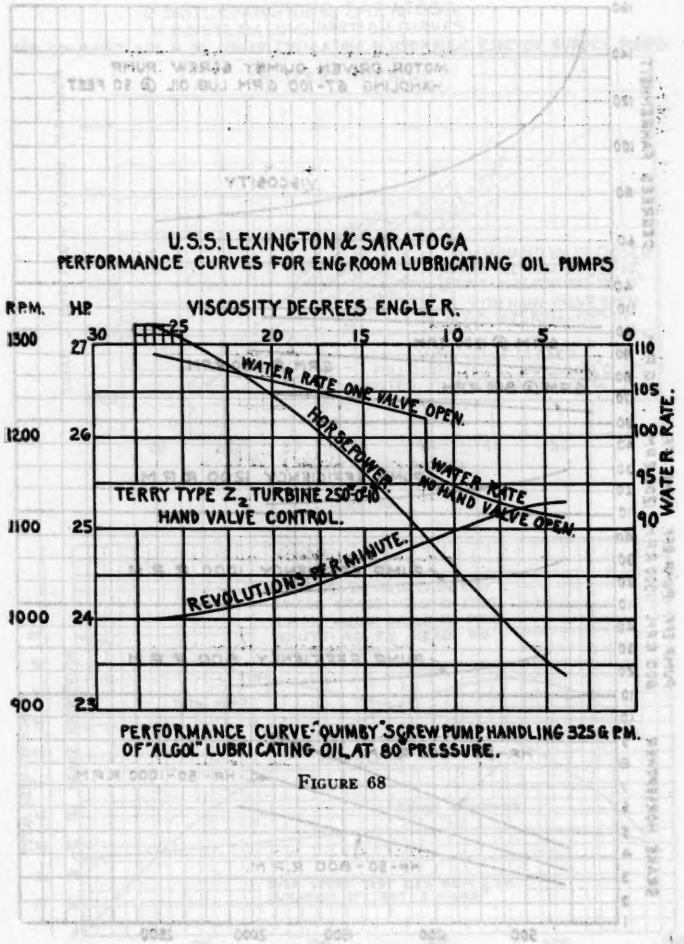

U.S.S. LEXINGTON & SARATOGA
PERFORMANCE CURVES FOR ENG ROOM LUBRICATING OIL PUMPS
R.P.M.
H.P.
VISCOSITY DEGREES ENGLER.
30
25
20
15
10
5
0
1300
27
1200
26
1100
25
1000
24
900
23
110
105
100
95
90
WATER RATE.
WATER RATE ONE VALVE OPEN.
FOR BE POWER
WATER RATE
NO HAND VALVE OPEN.
TERRY TYPE Z2 TURBINE 250 B.H.P.
HAND VALVE CONTROL.
REVOLUTIONS PER MINUTE.
PERFORMANCE CURVE-QUIMBY SCREW PUMP HANDLING 325 G.P.M.
OF "ALGOL" LUBRICATING OIL AT 80 PRESSURE.

FIGURE 68

The load curves for the D.C. electric plants, Figures 30 and 31, and the performance curves for the fuel oil service and main feed water heaters, Figures 35 and 36, were obtained under operating conditions afloat. The remaining curves indexed in Table 12 were obtained from the manufacturers and the Naval Engineering Experiment Station tests.

CONCLUSION.

Because the *Lexington* and *Saratoga* have the highest powered machinery installations afloat at this writing, and inasmuch as high powered turbo-electric installations probably will be considered for future high-powered passenger liners, it will be of interest to note that the machinery weights under the cognizance of the Bureau of Engineering for each of these vessels total about 7500 tons, dry (8000 tons, wet). Of these weights, about 640 tons, dry (643 tons, wet), are non-propulsion engineering weights external to the compartments enclosing the boilers and main propelling machinery.

Since the usefulness of the data contained in an article of this type is dependent upon its accuracy, care has been taken to present the tabular data as observed and to check the calculated results.

ALUMINUM FOIL HEAT INSULATION.

By Lieut. Comdr. T. A. Solberg, U. S. N., Member,

AND

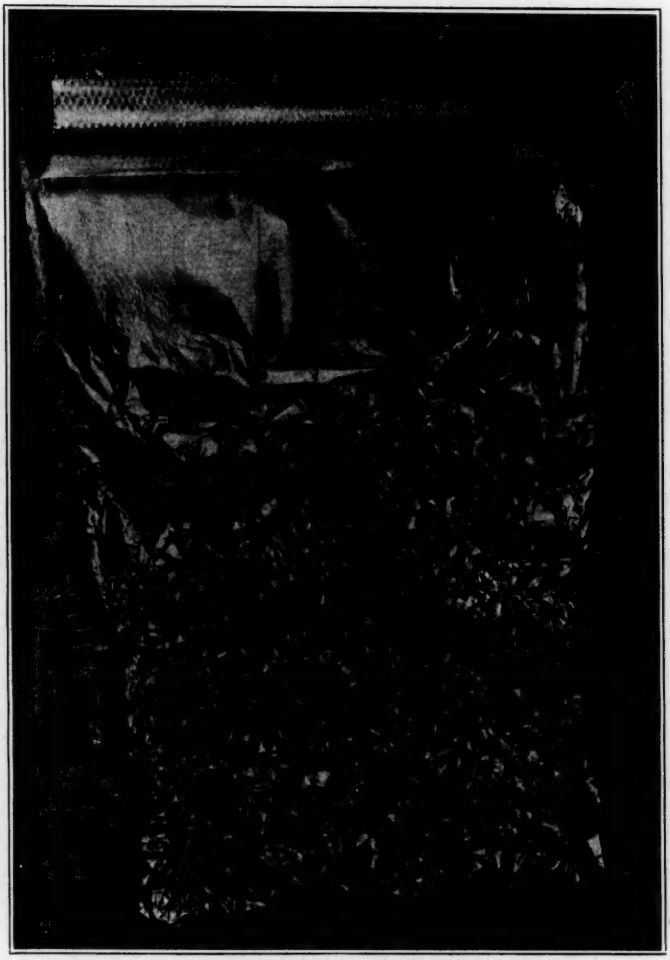
Mr. W. P. Sinclair.*

The many and diversified uses of aluminum make it perhaps the most versatile of all metals in modern industry. One of the recent uses to which aluminum has been put is in the field of heat insulation.

At first glance the idea of using a metal as an insulating material to retard the flow of heat seems foolhardy. Nevertheless, very thin, pliable, sheets of aluminum foil can be applied in such a manner as to form an efficient heat insulating material. When so used it has heat insulating properties at least equal to those of the usual and more widely known materials, and in addition, has numerous advantages.

The use of aluminum foil as a heat insulator was first proposed in Germany by Professor Ernst Schmidt and Dr. Edward Dyckerhoff and patents were subsequently granted to them as of 1925. These patents covered methods for the application of thin sheets of the foil, approximately 0.0003 of an inch in thickness, either in plain or crumpled form. The plain foil was used and installed on straight lengths of pipe, applied in concentric layers, spaced 3/8 of an inch apart. The spacing was accomplished by using built up spacers or supporting rings. The next development was applying crumpled foil to fittings and pipe bends where the use of supporting rings was impracticable. For this application the sheets were crumpled by hand and applied as before in concentric layers. In this method each layer was made to occupy a space of 3/8 of an inch solely by crumpling the foil.

* Assistant Mechanical Engineer, U. S. Naval Engineering Experiment Station, Annapolis, Md.



Roll of Aluminum Foil Showing Plain and Crumpled Form.

The major insulating effectiveness of this material is derived from the character of the surface of the foil and the method of application. This results in providing a myriad of bright mirror-like surfaces which reflect back toward the source of heat, any waves which strike them. Also, since bright bodies are poor heat radiators, very little of any heat absorbed by them is passed on away from the heat source by radiation.

The application of the plain foil was laborious, costly and time consuming because of the necessity of providing supports for each layer. Since it was found that satisfactory results were obtained with the crumpled foil on fittings and bends, the original method was discarded in its favor. This method provides a more substantial and rugged installation which is not so susceptible to damage.

To apply the material in this way, the foil is crumpled by hand so that the sheet presents a series of small mounds which have sharp points or edges as their apexes. Consequently when one layer is applied over the other the metallic contact between the layers is reduced to a minimum. At the same time this produces a very large number of confined air pockets. Any number of layers of foil desired can be used depending on the size and shape of the surface to be covered, the temperature, and the results desired. It has been found that the best and most economical results are obtained as stated above, when each layer of the crumpled foil occupies a space of ⅜ of an inch. In general the total number of layers required for comparable efficiency with other insulating materials is that which will give an overall thickness equal to that of the other material. For instance four layers of the crumpled foil gives about the same efficiency as 1½ inches of other types of pipe covering.

The insulating efficiency of this material is dependent on three factors, namely—the large number of self contained air spaces formed between successive layers of the foil, the small amount of metallic contact between layers, and the heat reflecting properties of the numerous bright metallic surfaces in each layer. It is a well-known fact that the greatest heat loss through dead or confined air spaces occurs by radiation. It is also well established that any bright metallic surface has high heat reflecting proper-

ties and conversely, poor heat absorption values. Hence it is easily seen that the major loss which can occur, is reduced to a minimum by the action of the great number of bright surfaces present in each layer, whereby most of the heat radiated to each successive layer is reflected back in the direction of its source. Consequently the first layer reflects directly back to the insulated surface most of the heat reaching it and at the same time radiates but a small amount in the direction of the next layer of foil. Each succeeding layer of foil, therefore, receives considerably less heat than the one next nearer the high temperature source.

The next greatest loss of heat through an insulation of this type is that from conduction. As previously explained, the crumpling produces only short line or edge contacts between adjacent layers. In order to still further reduce the amount of metallic content, the surface of the metal in rolling, is embossed with small corrugated diamond shaped figures, dots, irregular figures or shapes. This treatment has a surprisingly beneficial effect on the ease of crumpling in addition to providing the raised or uneven surfaces which result in minimum contact with contiguous sheets and hence low conduction of heat. This also results in an increase in the number of reflecting surfaces.

The aluminum foil for use in this work is commercially pure aluminum and is supplied in long sheets wrapped on hollow metal or fiber rolls so that it can be easily unrolled. It is supplied in a thickness of about .00045 of an inch and in various widths from 12 to 16 inches. A single roll of the foil 7 inches in outside diameter on a 1-inch diameter tube center carries a length of about 3700 feet and the material weighs only 12 pounds. Based on a width of 12 inches, this makes 3700 square feet of foil, which when crumpled makes up into approximately 800 board feet of applied insulation.

In applying this type of insulating material to the work some difficulties may at first be encountered but these, with a little experience, are easily surmounted. Although the crumpling must be done carefully by hand it can be done quickly. The material has a fairly high tensile strength but tearing and splitting must be guarded against. The sheet is crumpled as it is unrolled, is then cut into any desired lengths and individually applied. For the

purpose of sealing and thus preventing the formation of air currents, the edges of each sheet are brought together and folded over together. Also for the same purpose the layers are overlapped generously and the joints of successive layers staggered with those beneath.

In view of the nature of the material, each application, whether on pipe or flat surface, must generally be protected by an outside sheathing of thin metal such as galvanized iron or aluminum. But in out of the way or safe locations no such covering should be required. Special construction must be provided for the sheathing. For piping or curved surfaces, a series of stiff wire rings, each supported by radial sheet iron posts or prongs secured to the pipe with metal straps, forms the supports. For flat surfaces the sheathing can be secured to horizontal or vertical supports in between which the foil has been applied.

Although comparatively little experience has been obtained with the foil type of insulation in this country, reports show that it has been used successfully in Germany, Denmark, France and Italy in power plant, railway and marine service. Its usage has also been extended to the insulation of buildings, railway refrigerator cars, and in passenger cars as a protection against temperature changes of the weather. For many of these applications the sheets of foil properly spaced can be simply suspended from racks.

The durability or lasting qualities of the aluminum foil has been questioned. Rapid oxidation and disintegration would make it short lived and hence undesirable. The results of tests, however, have shown that because of the peculiar characteristics of the metal, the foil loses only about one per cent of its initial reflectivity from exposure to normal atmospheric conditions. In a standard 3 per cent salt water spray test, corrosion spots appeared on the foil after 40 hours, and after 200 hours small holes were present. This, however, is a severe test and indicates that the foil has considerable resistance to corrosion.

In order to increase the resistance to corrosion, a foil coated with a special lacquer can be used. This lacquered material in the same salt spray test failed to produce any tarnish, and after 200 hours in the spray box only a few minute holes were present.

In addition to having equal and very often better heat insulating properties than other insulating materials of the same overall thickness, when used for temperatures from 0 to 1000 degrees F., the aluminum foil has the following advantages:—

(1) Saving in weight:—The weight, even with protecting sheathing and its necessary supports, is generally less than that of other materials. This saving increased with the size and thickness of the insulation. In some cases it amounts to as much as 300 per cent. This feature makes it specially attractive for marine work.

(2) Cleanliness:—No dirt or dust results from work incident to installation or removal.

(3) Resistance to disintegration and "lumping" from vibration:—Because of its light weight and consequent low inertia force, vibration has little if any effect on the foil, and therefore it tends to remain in place as installed.

(4) Impervious to water and moisture:—Even though not protected by sheathing or metal covering, water does not penetrate easily into the mass of the material. Its resistance to corrosion is good.

(5) Low weight and space requirement of the material before application:—This results not only in ease of transportation and packing, but enables carrying a stock of material for replacement and repair in a minimum of storage space. Handling and transportation charges are obviously low.

(6) Low heat storage:—Because aluminum has a low specific heat and only a small weight of it is required, it follows that small heat losses occur, incident to warming up operations and, consequently, less time is required for this procedure. This is also of value where intermittent operation is concerned.

(7) Ease of Manufacture:—The manufacturing process is simple and requires a minimum of machinery and labor.

(8) Fireproof.

(9) Range of application:—The material shows insulating efficiencies in the neighborhood of 95 per cent over the wide range from 0 to 1000 degrees F. Few if any other types of material

are suitable for application, over such a wide range of temperatures.

In conclusion it might be stated that the disadvantages of this material for heat insulating purposes are relatively few. The principal difficulty encountered so far seems to be the requirement of thin stiff metal sheathing for outside protection in so many applications. This sheathing in turn requires special provision for its support and securing. Improvement in the present methods, however, may remove this difficulty. In spaces where the sheathing is not necessary the installation is simplicity itself. Another point to be considered is that the effectiveness of the insulation will be partially lost if the inner layers lose their reflecting power. A small amount of microscopic tarnish of the layers has shown only a slight change in heat insulating properties. Also, from data obtained to date, it appears that this condition is not likely to develop within a reasonable length of time. This condition, of course, is not apt to arise in the case of the foil which has been treated with a thin coat of lacquer.

THE TREND IN NAVAL ENGINEERING.

Aviator's Goggles, Electric Switches, Scaling Hammers, Vegetable Peelers and Substitute for Brass.

By C. Huey, Civil Member.*

Note by Editor:—This is the seventh of a series of articles on "The Trend in Naval Engineering," the first of which was published in the February, 1931, number of the Journal.

1. In the last issue of the Journal, it was stated that there were various accessories or devices used in Naval service which are little known, but which have served to help in the evolution to higher efficiencies, quicker results, saving in time, saving in weight, or saving in manpower. The scope of this article covers five additional selected items of general marine interest, which will be discussed in terms of their principal functions and trend in design.

AVIATOR'S GOGGLES.

2. One of the essential items for the safety, comfort and efficiency of the pilots and observers of open cock-pit aeroplanes such as are used for military purposes, is a properly designed and fabricated goggle. The desirable qualifications for a satisfactory goggle in order of importance are, undistorted vision, wide angle of vision, freedom from fogging, comfort, and resistance to breakage from flying particles.

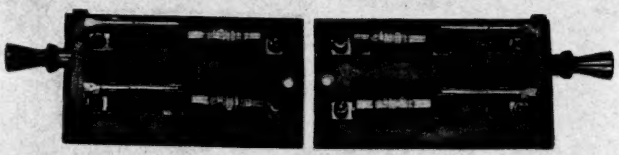
3. In the earlier days before laboratory tests were incorporated in the purchase specifications, the desirable qualifications of the goggles were in evidence only to the extent that they were the best the market afforded. (See Plate 1.) They were as a matter of fact merely uncomfortable, fragile, curved or flat windshields, with various amounts of prism, focus and astigmatism. Normal eyes require only optically plane lenses. Defective eyes require corrective lenses ground to a prescription formula based on the individual requirements. The presence in a pair of lenses of unknown and unprescribed prism, focus or astigmatism, results in misjudg-

* Material Engineer, U. S. Navy Yard, New York, N. Y.

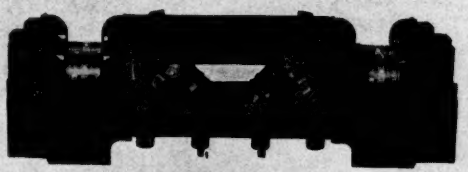
OLD WAY
PLATE 1.

NEW WAY
PLATE 2.

AVIATOR'S GOGGLES.



OLD WAY
PLATE 3.



NEW WAY
PLATE 4.

ELECTRIC SWITCHES!

ment of distances, and produces eye distortions known as esaphoria (converging eyesight), hyperphoria (diverging eyesight), cyclaphoria (rotation of eyesight), myopia (near sightedness), hyperopia (far sightedness), or astigmatism (myopia or hyperopia in a single plane).

4. The latest design of aviator's goggles, known as the Navy type, is shown in Plate No. 2. These consist of a molded, velvety, sponge rubber, chamois lined, air-tight face pad, of size and form to comfortably bear on the forehead and face. The pad has appreciable bearing surfaces which serve to distribute the pressure of the goggle resulting from the wind or tension of the flexible headband. The sensitive nerves and blood vessels in the vicinity of the eyebrows are avoided by the pad, so that comfort is obtained when wearing the goggle for flights of long duration. The pad and lens frames are sufficiently large to permit the wearing of personal corrective spectacles. In order to obtain a wide angle of vision, both in the horizontal and vertical planes, and to obtain optical characteristics which would combine practical freedom from prism, focus and astigmatism at the pupilary center of the lens, the spherical lens, known optically as a de-centered compensated miniscus lens, was adopted. These lenses are optically ground to a formula with precision accuracy. The acceptance tests on these lenses are very exacting and are made by viewing the lenses through a focal indicating, 20 power telescope, directed at a clock target 50 feet distant. The slightest optical errors are readily detected and evaluated by this test.

5. The mechanical strength of the lenses, in their resistance to fracture when struck with hard objects, is to a certain extent inherent in their spherical shape. This feature, however, has been greatly increased by case-hardening the glass so that it resists fracture when struck with a 5/8-inch diameter steel ball, dropped 36 inches on the center of the supported lens.

6. Freedom from fogging on the inner lens surface of the Navy goggle is obtained by an evacuated air system produced by the suction action of a Pitot tube in the slip stream of the goggle. This device is attached neatly to, and integral with the top of the lens frame.

7. In addition to clear lenses, the Navy goggles can be provided with interchangeable filter or non-glare lenses of optical qualities equal to the clear lenses. The filtering qualities of the filter lenses are such that the harmful invisible rays are reduced to an inappreciable amount, and the visible rays are transmitted in sufficient amount to permit observation when spotting in the sun, or maneuvering in searchlight beams during night landings. The color of the filter lenses is blue-green in appearance. This shade gives maximum visibility at the wave-lengths most efficient to the human eye. The filter lenses, however, are non-color selective to the observer, which feature permits ready identification of signal flags, colored flares or color scheme camouflaging. Prior to the adoption of the non-color selective filter lenses, amber and didymium glass were used. These materials are color selective which made it possible, with pre-knowledge, for enemy camouflaging in selective colors to increase the factor of invisibility.

ELECTRIC SWITCHES.

8. A large number of electric switches are required on shipboard, to connect or disconnect the power supply to the various sections and compartments, for motor driven devices, indicating and control systems, electric illumination, etc. Wherever automatic overload circuit breakers were not essential, double pole, lever type, knife blade switches with enclosed fuses were used until recently, for this purpose. (See Plate 3.)

9. The objection to the use of the lever form of switch is the large amount of space required between opposite poles, and between the points to be disconnected. This is due to the possibility of destructive arcing when slowly opening the switch under load. To allow sufficient air gap distances for this requirement, and to allow for the operator's insulated handle, not only is considerable space required, but the necessary linear dimensions of the copper conducting parts results in excessive weight of metal. In addition, assemblies of lever type switches are required to be mounted on heavy insulating panels which in turn are supported on heavy steel frames. This procedure requires many drilled holes with consequent weakening of the panels.

10. Due to the large number of the smaller size switches rated 30, 60 or 100 amperes capacity which were required to be assembled on panel boards, the factors of space and weight becomes of increasing concern. Commercially, the space factor plus the exposure of the operator to live parts, resulted in the general use of enclosed snap or toggle switches in the 5 and 10 ampere sizes. For the Navy, the need for 30, 60, 100 and larger ratings, required the development of toggle switches suitable for the severe service on shipboard. One of the new designs of a two-circuit, fused, double pole, snap or toggle switch is shown in Plate 4.

11. In the above form of switch, the operator is not only protected from the live parts, but the space and weight factors are greatly reduced by virtue of the high speed arc rupturing mechanism, the speed of which is not dependent on the operator's manipulation. By the use of molded electrical insulation for supports, arcing barriers and protection to the operator, the completely assembled switches can be closely grouped in large numbers on skeleton metal switchboards, with resulting compactness and saving in cable connections.

SCALING HAMMERS.

12. Since the days when iron and steel replaced wood for hulls, bulkheads, deck plates, etc., or since the advent of power driven ships which requires steel smoke pipes, ventilator cowls, boilers, fuel compartments, etc., or since the conception of fighting ships with turrets, spotting tops, searchlight platforms, etc., one of the most common tasks performed on shipboard is the laborious removal, by hand labor, of blistered paint and rust scales. Very little headway has been made in the development of rustless metals for major applications. Likewise, the best known protective paints have a limited life under marine conditions.

13. The use of an efficient tool to chip away the paint and rust scales is not only a labor and time saver, but it increases the prospect of applying an efficient coating of paint if the metal surfaces are clean and smooth.

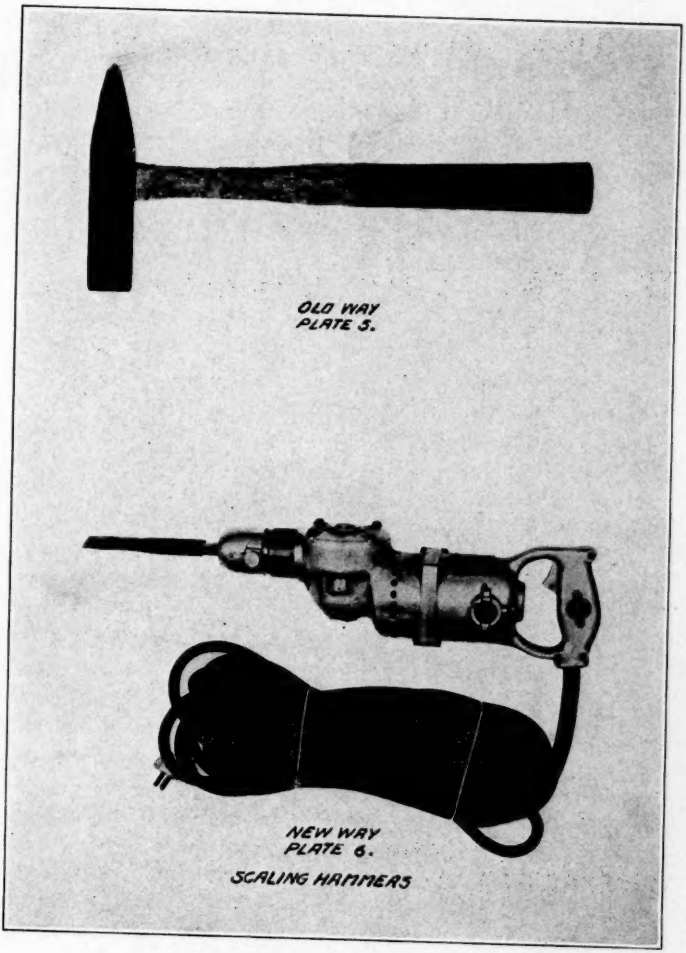
14. The common hand tool for this class of work is shown in Plate No. 5. The number of strokes per minute when using this

hand tool is, of course, dependent on the skill and " frame of mind" of the operator. This will vary probably from 60 to 170 strokes per minute with the average perhaps of 120. The power or force of the blow per stroke is likewise a matter of the operator. This will vary from ¾ to 1½ equivalent foot-pounds of work. Damage to the base metal is to be expected if the blow exceeds about one foot-pound. The effectiveness of the successive blows, of course, is dependent on their being properly guided by the operator.

15. There has recently been placed on the market several types of electrically driven scaling hammers which will supersede the hand tool for all purposes where time and efficiency is a factor. One of these types is shown in Plate No. 6. This consists of a high speed universal current motor of about 1/6 horsepower at 110 volts. The armature or rotor of the motor is geared to a crank-shaft which in turn operates an eccentrically mounted roller pin engaging with a slot in a reciprocating plunger. When the maximum momentum is imparted to the plunger, it is automatically released, striking a blow on a hardened steel disc. The disc is free to move ⅛ inch during the transmittal of the blow to a removable scaling tool. The complete device weighs about 12 pounds and is shaped to a practical form for convenience in handling. The force of the blow is equivalent to about ¾ of a foot-pound of work, and the blows per minute is about 2300. These factors have been designed so that the scaling is accomplished as rapidly as the tool is normally moved along the work by the operator. It has been determined that if the number of blows is less than 2000 per minute, the operator would have to retard his movements to obtain the results. If the number of blows per minute is over 3600, it would result in a certain number of duplicated impacts not required by the scaling process. In view of the fact that the number or force of the blows is not dependent on the operator, scaling of 3 to 5 coats of paint may be accomplished with the electrically operated scaling tool, at the rate of about one square foot of surface per minute.

VEGETABLE PEELERS.

16. On all Naval vessels, regardless of size or type, one of the rarely deferred routine daily tasks is the peeling of large quantities of vegetables such as potatoes, carrots, turnips, parsnips or

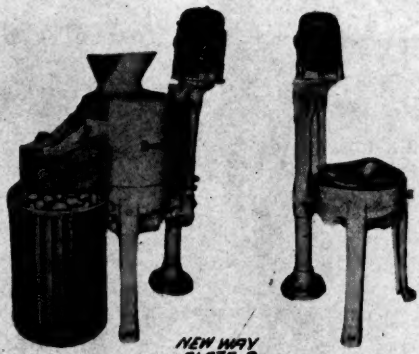


OLD WAY
PLATE 5.

NEW WAY
PLATE 6.

SCALING HAMMERS

OLD WAY
PLATE 7.



NEW WAY
PLATE 8

VEGETABLE PEELERS

sweet potatoes. In the earlier days this work was accomplished by laborious hand labor which required many hours of time and numerous hands. (See Plate No. 7.)

17. From the standpoint of economy, the hand process is not only wasteful in the use of able-bodied enlisted men, but the material waste from excessive peeling runs from 20 to 40 per cent, depending on the experience or "frame of mind" of the men assigned to this task. Excessive material waste of fixed allowances of vegetables per man results in reduction of the corresponding amount of food available. From another angle, if the rations are sufficient after allowing for excessive waste, then the saving from economical peeling could be applied to reduction in cost and weight of the gross amount of vegetables carried as stores. From another standpoint, hand-peeled vegetables, especially potatoes, usually stand too long while awaiting completion of the batch. This results in fermentation and deterioration at the exposed surfaces. In addition, modern sanitary requirements do not permit the promiscuous contact of the bare hands with the food while in the course of preparation for human consumption.

18. The modern method for peeling vegetables is by power driven machines which peel and wash the vegetable without contact with the hands, in batches of 25 to 50 pounds depending on the size of the machine. (See Plate No. 8.) These machines which are a part of every Naval ship equipment, consist of a cylindrical metal hopper lined with carborundum crystals fused into the surface of the metal. A metal disc of similar surface is located at the bottom of the hopper and is revolved at the proper speed by an electric motor. The contour of the top surface of the disc is designed so that when the hopper is filled to its capacity with vegetables, the individual vegetables are agitated and uniformly peeled on all surfaces by the abrasion produced by sliding contact with the disc and sides of the hopper. During the peeling process, which lasts only about one minute, the dirt and peelings are washed down a large drain by the action of a water spray.

19. Vegetables, especially potatoes, which are bruised, turn black on standing. On properly designed peeling machines, bruising is avoided by virtue of the special shape and suitable speed of the

revolving abrasive disc. Also the tendency to grind flat sides on the vegetables, is reduced to a minimum by proper agitation.

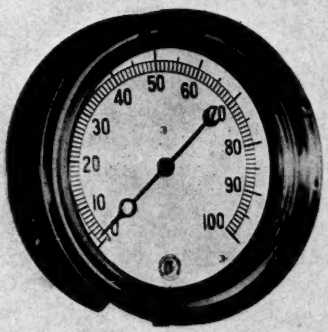
20. At the conclusion of the peeling, which is based on the operator's opinion of the results, or which may be done by automatic power shut-off, set to a pre-determined time interval, the vegetables are discharged from the hopper into a receptacle by opening the door at the side. The peeling losses of potatoes in the modern machine are consistently of the order of 8 to 12 per cent of the gross weight, depending on the age, size and quality of the potatoes.

SUBSTITUTE FOR BRASS.

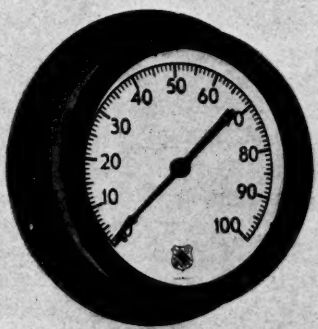
21. The operation of modern Naval ships requires hundreds of measuring and indicating devices such as pressure gages, speed indicators, annunciators, recorders, etc. The combined weight of these totals a substantial amount especially when the enclosing case is made of one of the heavier metals such as brass. The use of this metal has, for the past decade, been accepted on the basis as being the usual marine engineering practice.

22. Pressure gages have invariably been made of brass, either plain polished or nickel plated. The choice of brass no doubt was made on the basis of its resistance to corrosion combined with the artistic aspect. This latter factor has played more than a minor part in the selection of materials for accessories in Naval service as well as in universal commercial applications. The present trend in Naval ships as to the limitation in weights has served to throw the spotlight on many devices on shipboard which previously have been accepted as being satisfactorily developed for the purpose intended. The recent development of corrosion resistant, cast aluminum alloys has supplied a lightweight substitute for brass fairly acceptable for most of the applications of enclosures for the many devices.

23. The chief virtue of aluminum alloys, however, is their light weight. They do not excel in physical strength, elasticity or artistic appearance, and their resistance to corrosion is not of high order unless re-inforced with lacquer, paint or enamel. The electrolytic effect of aluminum in contact with dis-similar metals, is of



OLD WAY (BRASS)
PLATE 9.



NEW WAY (BAKELITE)
PLATE 10.

SUBSTITUTE FOR BRASS

serious amount and requires special knowledge in the application and choice of material for the securing screws, clamps, bolts, nuts or washers.

24. The undesirable characteristics of aluminum have served to react against using it for enclosing the mechanism of pressure gages. A recent development of a non-metallic, light weight, black material, known as high-strength bakelite (a phenolic condensation product) has suggested its application for pressure gages and similar devices where a non-corrodable, artistic, natural surface is desirable. This material is lighter than aluminum, and much lighter than bronze as indicated by their respective approximate specific gravities of 1.4, 2.7 and 8.5. The tensile strength, transverse strength, compressive strength and hardness, compares so favorable with that of aluminum and bronze, that little if any sectional changes are required in the design. Enclosing cases made of high-strength bakelite require no machine work on polishing, as the unit parts, including the thread for screwing on the bezel, are molded and cured to finished dimensions.

25. Plate No. 9 shows a Navy type six-inch diameter pressure gage with a cast brass case and bezel. The complete gage weighs 5 pounds 8 ounces. Plate No. 10 shows an equivalent Navy type gage with a high-strength bakelite case and bezel. This complete gage weighs 2 pounds 5 ounces. Bakelite for electrical instrument cases is being used in increasing amounts wherever iron cases are not required for magnetic shielding. Bakelite in these applications not only furnishes a lightweight non-corrodable finish, but due to its electrical properties, serves to enclose the mechanism in a high-grade, non-absorbent insulation.

ALKALINE STORAGE BATTERIES.

By F. Brehme.*

Storage batteries find their widest application in automobile starting and lighting service and it is through this channel that most people have their first experience with batteries. Because of its characteristics, the alkaline battery is rarely used for this service and it is not until an individual has need for batteries in other types of duty that he may learn of, and perhaps use, an alkaline battery. Consequently, knowledge of this type is not as general as of the lead-acid types and the user is sometimes confused as to the characteristics, application, and operation of an alkaline battery. It is the purpose of this paper to discuss the major features of the Edison alkaline storage battery, this being the only alkaline battery of commercial importance in use in this country.

CONSTRUCTION.

The positive active material is an oxide of nickel contained in perforated steel tubes and the negative active material is finely divided iron of exceptional purity, contained in perforated steel pockets. The positive tubes are rigidly assembled in a steel grid to form the positive plate while the negative pockets are likewise assembled in a steel grid to make up the negative plate. To form the positive group the positive plates are suspended from and connected by a steel connecting rod, as are the negative plates; the two groups are intermeshed so that the plates alternate positive and negative, the plates being separated by hard rubber pin insulators. The container is steel, rectangular in shape, with all seams welded. The pole pieces are steel, tapered, and threaded at the top; intercell connections are made by slipping the connector down over the pole piece and tightening a nut.

* Edison Storage Battery Co., Orange, N. J.

All of the steel is nickel-plated. The electrolyte is a potassium hydroxide solution with lithia. The steel construction results in great strength and durability combined with light weight.

THEORY.

Although the chemical actions which take place through a cycle of discharge and charge are complicated, the net result is a transfer of oxygen from the positive plates to the negative during discharge and a return of this oxygen during charge. When charged, the positive active material is a high oxide of nickel while the negative material is metallic iron; during discharge the high oxide reduces to a lower oxide and the negative becomes an oxide of iron. Neither the nickel nor the iron leaves the plates to combine with the solution nor does the potassium in the solution combine with the active materials in the plates. The electro-chemical action is completely reversible; with continued use of the cell there is no reduction of the quantity of active material in the plates and there is no deposit of active material at the bottom of the cell.

TYPES AND SIZES.

B Type cells range in size from 18¾ ampere hours capacity to 112½; A Type cells from 150 to 600 ampere hours; C Type cells from 225 to 675 ampere hours capacity. These three types have the same general plate construction but the plates are of different sizes. C Type cells are taller than A Type and, where vertical space is available, the use of C Type cells in place of A Type results in 50 per cent greater ampere hour capacity for the same area occupied.

G Type cells have slightly different plate construction with lower internal resistance; sizes range from 100 to 450 ampere hours capacity.

WEIGHT AND SPACE.

Weights and dimensions per kilowatt hour for alkaline batteries will vary with size but are approximately as follows:

Type.	*Lbs. per Kw. Hr.*	*Area per Kw. Hr., Square Inches.*	*Height, Inches.*
B	105	240	10¾
A	87	105	16–17
C	87	70	21¾
G	90	110	16–17

The space requirements for alkaline batteries are usually about the same or slightly more than that for lead-acid. The weight of alkaline batteries for any given installation will usually be from 17 per cent to 30 per cent less than that of lead-acid batteries as these weigh from 105 pounds to 130 pounds per kilowatt hour depending on size, make, and type of construction.

APPLICATION.

The classes of service where alkaline storage batteries are applicable include electric industrial trucks and tractors, electric street trucks, storage battery locomotives, emergency lighting, marine auxiliary power, oil circuit breaker control, radio, railway signal circuits, alarm circuits, railway car lighting, and certain types of telephone circuits. Those classes of service where the use of alkaline storage batteries is not recommended are as follows:

Submarine: Alkaline batteries are not manufactured in capacities large enough to meet submarine requirements. Although a few experimental batteries were built about 1915, there have been none made since that time.

Central Station: The same limitation applies. Sizes available are too small for power house standby service.

Automobile, Marine, or Airplane Starting and Lighting: The characteristics of the alkaline battery are such as to make it less suitable for the type of duty required in these services than the lead-acid type, and long life is here of minor importance.

Telephone: Alkaline cells are not made in capacities of sufficient size for most large telephone installations. However, in some of the smaller installations, alkaline storage batteries can be used satisfactorily; this being a matter of individual circuit arrangements.

LIFE.

The life of alkaline cells is guaranteed variously from 7 to 16 years depending on the class of service in which they are to be used. Those batteries which are properly applied and receive good care usually outlive the period of the guarantee.

DISCHARGE CHARACTERISTICS.

Alkaline cells are rated 1.2 average volts per cell to a final voltage of 1.0 volts per cell at the 5 hour rate for A, B, and C Type cells and at the 3⅓ hour rate for G Type cells. However, not only do all batteries diminish in voltage as the discharge progresses but the rate of discharge affects the voltage; the higher the discharge rate the lower will be the voltage. Consequently, when a manufacturer wishes to state the average voltage, final voltage, and ampere hour capacity to be obtained from a cell, it is necessary to arbitrarily select both an ampere rate of discharge and a final voltage limit. In services where alkaline batteries are used, lead-acid batteries are rated variously at the 6, 6½, 8 and 10 hour rate. Although the ampere hour capacity of the alkaline battery is affected only slightly by various rates of continuous discharge, the ampere hour capacity of lead-acid batteries is very seriously affected and this must also be taken into consideration.

In making comparisons between alkaline cells and lead-acid cells, then, such comparisons should be on a comparable basis as to the rate of discharge and as to the ampere hours delivered. As an illustration, below is given tabulated comparisons of typical A Type and G Type alkaline batteries as against typical lead-acid batteries of approximately the same kilowatt hour rating. The tabulations cover various rates of continuous discharge, the capacity delivered is that capacity for each different rate where the discharge of the lead-acid battery should be terminated to avoid injury to the battery, as recommended by the manufacturer. The discharge of the alkaline battery may be continued beyond this point without injury but, of course, the voltage will continue to drop until eventually the voltage is too low to be of value; the useful limiting voltage in any particular case will be dependent on the requirements of the equipment being served by the battery. However, discharging the alkaline battery all the way to zero voltage will not injure it.

"A" TYPE ALKALINE BATTERY VS. LEAD-ACID BATTERY.

Alkaline: 160 cells, 57.6 Kw. Hours (at 5 hr. rate).
Lead-acid: 100 cells, 60.3 Kw. Hours (at 6 hr. rate).

	Discharge Rate	*Capacity* Delivered*	*Initial† Voltage*	*Final Voltage*
Alkaline	50 amps.	312 amp. hrs.	220	179
Lead-acid	50 amps.	312 amp. hrs.	204	176
Alkaline	100 amps.	245 amp. hrs.	207	177
Lead-acid	100 amps.	245 amp. hrs.	201	173
Alkaline	150 amps.	206 amp. hrs.	195	167
Lead-acid	150 amps.	206 amp. hrs.	198	169
Alkaline	200 amps.	185 amp. hrs.	183	157
Lead-acid	200 amps.	185 amp. hrs.	195	165

"G" TYPE ALKALINE BATTERY VS. LEAD-ACID BATTERY.

Alkaline: 160 cells, 52.6 Kw. Hours (at 3 1/3 hr. rate).
Lead-acid: 100 cells, 52.8 Kw. Hours (at 6 hr. rate).

	Discharge Rate	*Capacity* Delivered*	*Initial† Voltage*	*Final Voltage*
Alkaline	50 amps.	274 amp. hrs.	224	188
Lead-acid	50 amps.	274 amp. hrs.	204	176
Alkaline	100 amps.	215 amp. hrs.	214	186
Lead-acid	100 amps.	215 amp. hrs.	200	172
Alkaline	150 amps.	181 amp. hrs.	207	179
Lead-acid	150 amps.	181 amp. hrs.	197	169
Alkaline	200 amps.	156 amp. hrs.	196	171
Lead-acid	200 amps.	156 amp. hrs.	193	165
Alkaline	250 amps.	140 amp. hrs.	188	162
Lead-acid	250 amps.	140 amp. hrs.	190	161
Alkaline	300 amps.	129 amp. hrs.	178	156
Lead-acid	300 amps.	129 amp. hrs.	186	157

* Limiting capacity for lead-acid batteries at given discharge rate.

† Initial voltage as given for alkaline batteries is at 2 per cent rated capacity out. Initial voltages at the beginning of discharge will be slightly higher for a few minutes until 2 per cent of the rated capacity has been used: the voltage during this brief period will depend on how long the battery stands idle after being charged. A fully charged alkaline battery has a small amount of unstable oxygen in the positive plates which gradually escapes in the form of a gas when the battery stands idle. If the battery is used before any of this unstable oxygen has worked off, the initial voltage may be as much as 7 per cent higher at the 5-hour rate. The additional voltage obtained thereby is extra and beyond the rated average voltage of the cells.

CHARGING.

In cycle charging, that is, where a battery is alternately charged and discharged such as in industrial truck service, alkaline batteries may be charged by either the Constant Current method or by the Modified Constant Potential method. In either case, it is necessary to put back into the battery approximately 25 per cent more ampere hours than have been used on discharge; if ampere-hour meters are used then these should be set accordingly; where ampere-hour meters are not used it is recommended that the batteries be charged until the terminal voltage, with constant current flowing, ceases to rise over a period of 30 minutes. Where all or most of the capacity has been delivered on discharge, the charging time will be from 6 to 7 hours.

Constant Current charging requires a line voltage of about 1.8 volts per cell to complete the charge; there should be suitable variable resistance in series with the battery so that the charge may be started at approximately 1.6 volts per cell. For A, B, and C Type cells it is recommended that the charge rate be held at approximately ⅕ of the ampere hour rating of the battery, that is, a 150 ampere hour capacity battery would be charged at 30 amperes; for G Type cells, the charging amperes should be held at about ⅓ of the ampere hour rating.

Modified Constant Potential charging requires a line voltage of at least 1.84 volts per cell with a fixed resistance in series with the battery equal to: Line Voltage minus (1.7 times Number of Cells) divided by Ampere Hour Rating of Battery times 5 (for A, B, and C Types) or times 3 (for G Type).

In continuous charging, that is, where a battery is used intermittently but is otherwise on charge such as in emergency lighting and other standby services, it may be Trickle Charged or Floated.

To Trickle Charge (low rate charging) a voltage of from 1.5 to 1.6 volts per cell is required. The rate of charge is determined as follows:

(Rated Ampere Hour Capacity times 0.16) plus (Average Ampere Hours Used Daily $\times$ 1.10) divided by 24.

To Float, that is, where the battery is constantly on the line and may be either charging or discharging according to line voltage

fluctuations caused by varying loads, the charging voltage required will be from 1.5 to 1.7 volts per cell depending on the charge rate required to keep the battery full, and the rate of charge will depend on the discharge demands for any given installation.

TEMPERATURE.

Extremes of electrolyte temperatures are to be avoided; however, in this connection electrolyte temperatures should not be confused with air temperatures. The charge and discharge of an alkaline cell under normal conditions will result in electrolyte temperatures or from 60 degrees F. to 105 degrees F.

Charging at high rates for long periods will cause high electrolyte temperatures and discharging at high rates for long periods when the battery is in a closed compartment will cause high electrolyte temperatures. Temperature should be kept below 115 degrees F.; adequate ventilation of the battery compartment aids materially to prevent excessive temperatures.

Low electrolyte temperatures will result in temporary loss in capacity and where the electrolyte temperature remains below 40 degrees F. the loss is important. However, even though the electrolyte temperature be below 40 degrees F. at the start of discharge, the working of the cell will gradually raise the temperature to a satisfactory value in most cases. Where batteries are to be exposed to a very low air temperature for any length of time and operation is intermittent it is sometimes advisable to insulate the battery compartment, this results in more rapid heating of the electrolyte on discharge and it also results in longer retention of warmth when the battery stands idle.

GASSING.

When being charged, all batteries give off hydrogen and oxygen gas in an explosive mixture; alkaline batteries gas slightly more than lead-acid batteries. Where charging is done in a closed battery room there should be provisions for ventilation. Also, tools used around the battery should be handled carefully to avoid short circuiting across cell terminals and thus causing an arc to ignite the gases escaping from the battery, the gases are most concentrated immediately above the cells.

When being discharged, the gassing of an alkaline battery is negligible unless the discharge is carried out to a point where cell voltage goes below zero volts and the cells are thereby reversed. Under ordinary conditions of operation this should not occur but when it does the ventilation provided for charging is more than adequate; under such a condition the same care should be used in avoiding arcing at the cell terminals.

INSTALLATION.

Inasmuch as alkaline batteries do not give off fumes which are corrosive to iron, copper, steel, etc., they may be installed in close proximity to other equipment; no special battery room is required. It is not necessary to sand the floor but the surface on which the trays rest should be clean and dry, if there is a tendency to dampness it is wise to use glass or porcelain insulators under the trays.

When making connections between trays be sure that the pole nuts are tightened down firmly to assure good electrical contact.

MAINTENANCE.

The main points in caring for alkaline batteries are as follows:

1. Maintain solution level at the proper level, as recommended in the manufacturer's instructions.
2. Avoid electrolyte temperatures exceeding 115 degrees F.
3. Keep clean and dry the cells, trays, and the surface on which the trays are set.

DISCUSSION.

SOME COMPARISONS IN PROPELLER DESIGN METHODS.

A Criticism by Commander J. M. Irish, U. S. N.

Commander Davis' article in the February issue of the A. S. N. E. Journal will be very useful to those who desire to check their methods by using more than one of the standard systems of propeller design, but to the careful reader it emphasizes the following conclusions:

(a) Design by any of the methods outlined is a difficult, complicated process.

(b) A large number of assumptions must be made.

(c) A wide variation in dimensions is possible, any of which apparently gives equally good propulsive results.

To use any of the methods described requires very considerable study and a wide experience on which to base the assumptions. In Froude's, Schaffran's, Taylor's, and Schmidt's methods the correct values for wake and thrust deduction factors are essential, yet these factors are not known and vary widely. Technical literature is full of the study of these variables and there is little assurance that much real progress is being made in their correct determination.

How to determine the best area is not described, but is assumed by Commander Davis. This is an important omission, as the correct area is essential to avoid cavitation.

In Dyson's method the correct determination of K and S is a study in itself. The number of variations to each rule and the exceptions to each case are so numerous that any value adopted is open to considerable question.

From the summary it will be found that the dimensions and S. H. P. vary between the following limits:

	Dyson	*Froude*	*Schaffran*
Pa/Da	.55	.44	.475
Diameter	9.17	10.26	9.47
Pitch	8.42	8.05	8.50
S. H. P.	26,500	25,000	27,850
R. P. M.	322	322	322
Pa □"	5,250	5,230	4,850

With such a variety of results to choose from, it would appear logical to assume that a great deal of latitude is permissible and that much of the involved discussions concerning propellers are probably " smoke screens" thrown off in an effort either to obscure the real cause of the differences or to increase the reputation of the various experts by making it seem difficult and involved.

In actual fact, the design of propellers can be made simple, direct, and accurate. Although Commander Davis did not consider it in his review, an article was published in the A. S. N. E. JOURNAL of May, 1930, describing such a method.* To demonstrate its easy use, the same problem as is done by Commander Davis in his review is worked by this method.

Given:

Number of shafts	4
Number of blades	3
Design speed	21.0 knots
Corresponding e. h. p.	15,740
Design r. p. m.	322

Solution: $C = \frac{\text{e. h. p.}}{N \times v^{3.1}} = \frac{15{,}740}{4 \times 12{,}600} = .311$

$$\text{r. p. m. per knot} = \frac{322}{21} \times 1.025 = 15.7$$

(Correction for 4 shaft ship = 1.025)

*From Sheet 1 — D plus P = 17.7

$$\text{Diameter coefficient} = \frac{\text{e. h. p.}}{N \times v \times R^2} = \frac{15{,}740}{4 \times 21.0 \times (322)^2} = 0.0018$$

Diameter (est.) 9′.6 — from Sheet IV*

$\therefore$ P = 17′.7 — 9′.6 = 8′.1

*From Sheet II, K = 18.5 (using C and [D + P])

(Correction for 4 shaft ship = .925)

$\therefore$ K = 18.5 × .925 = 17.1

$$\text{Total} = \text{S.H.P.} = 4 \times \left(\frac{\text{r. p. m.}}{K}\right)^3 = 4 \times \left(\frac{322}{17.1}\right)^3$$

$$= 4 \times 6680 = 26{,}700$$

To find area :

$$\text{T. S.} = \pi \times D \times R = \pi \times 9.5 \times 322 = 9620$$

$$Ts = \frac{\text{S.H.P.} \times 326}{N \times v} = 104{,}000$$

T/□″ limit for T. S. = 2.5 × .85 = 21.0 Sheet III*

(allowance for fouling and overload = .85)

$$\therefore \text{Pa} = \frac{104{,}000}{2.0} = 4950$$

$$\text{Pa/Da} = \frac{4950}{10{,}200} = .48$$

Final data : Pa/Da — .48

Dia. — 9.5

Pitch — 8.2

S.H.P.— 26,700

r.p. m.— 322

It will be noted that the design proceeds logically from given data to final dimensions, that no assumptions of any kind are made, that the only corrections are for transferring data from the sheets, which are prepared for twin screw vessels, to those for a four-shaft ship.

As justification for using the D + P method, it is interesting to find those given by the various designers:

	Taylor	*Schaffran*	*Froude*	*Schmidt*	*Dyson*	*Ave*
D + P	17.55	17.97	18.31	18.04	17.59	17.89

Compare this average with the figure 17.7 given above and the accuracy of the method cannot be doubted.

* "Simplified Propeller Design" by Commander J. M. Irish, U. S. N.

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NAVAL CONSTRUCTION IN 1931.

By HECTOR C. BYWATER, A.I.N.A.

No. I.

At the close of the first decade of naval construction governed by international treaty, it is both interesting and instructive to review the effects of this political innovation so far as they have developed. The purpose of the Washington Treaty was not merely to limit the dimensions of naval armaments, but to eradicate the spirit of competition which was inflating the defense budgets of the Great Powers and breeding mutual suspicion. To what extent, then, have these objects been achieved? So far as capital ships are concerned, the Treaty has been exceedingly effective, for it is nine years since the last vessels of this type were laid down. Until recently there seemed every prospect of the "battleship holiday" being extended to 1936, but this hope has been frustrated by the decision of France to resume the building of capital ships. Even so, the prolonged truce in heavy construction has resulted in the saving of immense sums of money, and that is a definite gain. But in other respects the system of treaty restriction has not fulfilled the hopes of its authors. It has burdened the various navies with ships of inordinate cost and problematic efficiency; it has not eliminated the competitive spirit, but only diverted it into other channels, and, above all, it has played such havoc with the balance of power that strategists and statesmen are left vainly groping for a yardstick with which to measure the relative strength of their own navy or that of any other.

Let us take these points in their sequence. Since the Washington Treaty came into force, the five signatory Powers have laid down fifty-two cruisers of the 10,000-ton 8-inch gun type, which represents the maxima of tonnage and gun caliber permissible for non-capital ships. Can it truthfully be asserted that one of these ships was built to perform any specific function of strategy or tactics other than those arising out of the creation of the type itself? We think not. A compromise between the light cruiser and the heavy armored cruiser, they exhibit the defects of each with few of the qualities of either. Their first cost is exorbitant—£2,000,000 per ship in this country—and they are very expensive to maintain, so that their multiplication is restricted quite as effectively by economic considerations as by treaty rules. The fighting value of these ships is most difficult to assess, owing to the wide discrepancy between their offensive and defensive powers. They are heavily armed, but have only the flimsiest protection, and it is generally accepted that a few well-aimed salvoes, or a single mine or torpedo, would be sufficient to disable, if not to destroy, them. They are certainly incapable of lying in line of battle, and to venture even momentarily within range of capital ships might be fatal to them. They are unduly large for the task of hunting down surface raiders, except when these are of their own type, and the same objection, added to extreme vulnerability, militates against their use as battle fleet scouts or as destroyer supports. They are, in short, hybrids for which it is difficult to find a place in the naval organization. Never really popular with naval officers, they are now almost universally discredited, and their formal suppression as a type would be greeted with general relief. Only in the United States do they still find advocates, who believe that the maintenance of the existing cruiser standard serves as a deterrent to excessive building.

That competition in naval armaments has survived the Washington Agreement is not difficult to prove. The imposition of a limit on tonnage and armament has spurred designers to get the utmost out of every ton at their

disposal. The American cruisers are remarkable for gun power; the French and Italian ships have phenomenal speeds, and the Japanese appear to combine both qualities at the expense of seaworthiness. Our own ships, of less spectacular design, are probably no better and no worse than their foreign contemporaries. The point is that there has been as much rivalry in the building of these 10,000-ton cruisers as there ever was in the construction of battleships, the only difference being that the participants in the new race have had less value for their money. In the production of lighter craft, competition is still very pronounced, France and Italy, for instance, being openly engaged in a building race for flotilla supremacy in the Mediterranean. In sum, therefore, the Washington Treaty has not proved an unmixed blessing, while the Treaty of London negotiated in 1930 is chiefly notable for the unilateral restrictions it places on British sea power.

THE BRITISH EMPIRE.

The cruiser *Exeter*, of 8400 tons, was the only large vessel to be completed during the year. Apart from minor modifications, she is a replica of the *York*. With their armament of six 8-inch guns and a speed of 32¼ knots, they are very useful ships, but the design is unimaginative and presents few points of interest. The installation of an auxiliary oil-motor plant for cruising purposes would have enhanced the utility of these ships by extending their radius of action. As they carry only 1900 tons of fuel oil, as compared with the 3200 tons capacity of the *County* ships of equivalent engine power, their sea-keeping endurance is very limited for their size. With the completion of the *Exeter* our quota of 8-inch gun cruisers is exhausted, and no further ship with this armament may be built before 1936. But, treaty or no treaty, it is not likely that we shall perpetuate either the *County* or the *York* class of cruiser, unless circumstances compel us to do so.

H.M.S. *Leander*, the solitary cruiser provided for in 1929, was launched at Devonport on September 25. She is the first of a new class, displacing 7000 tons and armed with eight 6-inch guns in turrets. These are the only official details so far released. It is understood, however, that a speed of 33 knots is anticipated, and that all the boiler uptakes are led into a single funnel. There is vertical armor protection over a limited section of the hull. Three further units of this class, authorized under the 1930 Estimates, are now on the stocks, the *Orion* at Devonport, the *Neptune* at Portsmouth, and the *Achilles* at Cammell Lairds, Birkenhead, and two more ships provided for in the current Estimates will be started this year. In view of the rapid shrinkage in our cruiser strength, owing to age and dilapidation, it will be necessary to build on a much larger scale during the years immediately ahead if we are to keep within measurable distance of the very modest establishment to which the London Treaty entitles us. Owing chiefly to the emasculation of the five-year program introduced in 1925, from which no less than eight ships have been dropped, and in part also to the delay in starting work on the annual programs, the cruiser position has become far from satisfactory. In the case of the *Leander* eighteen months intervened between the voting of the ship and the laying of her keel plate, while a similar delay occurred in starting the *Orion* and the *Neptune*. The last-named ships, together with *Achilles*—laid down in the early summer—form the cruiser quota for 1930, but they will not be ready for sea before the summer of 1934. Again, for the building program of 1931 so trifling a sum was voted in the Estimates that no serious work on any of the ships can be started until the late spring or early summer of next year, which means that they will be completed by the middle of 1935. Consequently the Naval Conference, which is due to be held early in 1936, will find Great Britain with

only six cruisers built and completed since the previous London Conference of 1930. This is an extraordinary position, the significance of which does not appear to be generally recognized. It means that we shall enter the next Conference almost destitute of the concrete bargaining assets which have notoriously swayed the balance at every disarmament congress in the past.

Of the three cruisers included in the current Navy Estimates, two are to be of the *Leander* class, of 7000 tons, while the third is to be a small vessel—approximately 5000 tons. A report that this vessel was designed for internal combustion machinery is officially discredited. The Admiralty evidently considers that the time is not yet ripe for effecting such a drastic revolution in the propulsion of cruising ships. To build a large motor warship at this juncture would be, no doubt, a rather bold experiment, difficult to justify when naval funds are so exiguous. In the absence of positive assurances we can but hope that the Admiralty is following the development of internal combustion machinery for naval use as closely as the importance of the subject deserves. Practical experiments are understood to be in progress at the Admiralty Laboratory with a new high-speed engine of dimensions approaching those of a light cruiser installation.

The flotilla leader *Kempenfelt*, built by J. Samuel White and Co., of East Cowes, and launched in October, is a duplicate of the *Keith*—Vickers-Armstrongs—which is already in service. They are leaders only by courtesy, for their displacement of 1330 tons is the same as that of the destroyers with which they will operate, and from which they differ only in having more cabin and bridge space and a lighter armament. The *Kempenfelt* is 323 feet overall, with a breadth of 32¼ feet, the machinery consisting of turbines with single reduction gearing and an output of 34,000 H.P. The main armament comprises three 4.7-inch guns. This vessel was authorized in 1929. The eight *C* class destroyers which were to have formed her flotilla were subsequently reduced to four, and all have been launched. Except that they mount four 4.7-inch guns, they are identical in essentials with the flotilla leader.

Only three submarines were voted in the 1929 Estimates. They represent a break in recent building policy, which has aimed at the production of large "overseas" boats of 1475 tons, of the *O*, *P*, and *R* groups. One unit of the 1929 group, *Thames*, is a minelayer of about 1760 tons; the other two, *Swordfish* and *Sturgeon*, are "coastals" of 650 tons, the smallest submarines we have built for many years. Size, however, is not a safe criterion of efficiency in the case of submarines, for in capable hands a small boat may be quite as formidable as a larger one. Moreover, the *Swordfish* is very strongly armed for her tonnage, six torpedo tubes being mounted in the bows alone. Three boats of this class can be built for the price of two "overseas" craft, and would probably represent better value. Three submarines authorized under the 1930 Estimates are on the stocks—the minelayer *Porpoise*, of the *Thames* class, at Vickers-Armstrong, Barrow; and the coastal boats *Starfish* and *Seahorse*, at Chatham Dockyard.

The "ca' canny" policy which has been pursued in recent years with reference to cruisers is equally marked in the case of submarines. Only thirteen boats have been authorized in the past four years, and three of them have yet to be laid down, while in the same period a larger number of older submarines have passed the limit of useful service. Since the war ended only twenty-five submarines have been provided for the British Navy, in contrast to the huge volume of submarine tonnage launched abroad. At the present moment, for example, forty-two submarines are under construction in France alone, and nearly thirty in Italy.

The *Bridgewater* class of small sloops, introduced in 1927, has now grown to fourteen units. They displace from 1040 to 1105 tons, have turbine machinery of 2000 S.H.P. for a speed of 16 to 16½ knots, and carry two 4-inch guns on high-angle mountings. These little ships have notoriously been built to eke out our shrinking cruiser tonnage, but actually their fighting value is negligible. The only function they could perform in war would be minesweeping. In size, armament and radius they are completely eclipsed by the French motorized sloops. It is a singular fact that one type of ship for which there would be an urgent demand in a national emergency has been studiously ignored in our post-war programs. This is the convoy cruiser, the development of which has been strongly urged by naval officers who were most intimately associated with the convoy system in 1917-18. The desiderata of this type are moderate speed, ample sea radius, and strong armament, all of which might be incorporated in a ship of less than 2000 tons. As ships below that tonnage are not limited by Treaty, there is nothing to prevent us from building convoy cruisers, which would be of incalculable value, in place of the present sloops, which are a dubious investment from any point of view.

UNITED STATES.

The last of the eight 10,000-ton cruisers authorized under the Act of December, 1924, was completed during the year. They are not uniform in design, for the *Pensacola* and *Salt Lake City*, which went afloat first, were discovered to be over-gunned and deficient in stability, and in the next ships the armament was cut down to reduce the weight on deck. Thus the *Augusta* and her five sisters differ from the *Pensacola* in mounting nine 8-inch guns instead of ten; the length has been increased from 585½ feet to 600 feet, and the breadth from 64 feet to 65 feet, and the freeboard has been raised. Although nominally of 10,000 tons, these American cruisers are officially claimed to displace only 9050 to 9100 tons, thanks to weight-saving methods of construction and the meticulous avoidance of cumbersome equipment. Apparently these methods have been carried too far, since a number of the ships have recently gone into dockyard with cracked stern-posts, while heavy weather and gun concussion have revealed other weaknesses. It is not denied that all eight ships are exceptionally lively in a seaway, a fact which appears to discount the practical value of their heavy armament. The propelling machinery is the same in all ships, consisting of Parsons geared turbines taking steam from eight White-Forster boilers—excepting in *Northampton*, which has eight of the Yarrow type—and developing 107,000 S.H.P. The mean trial speed was 33 knots. While this type may not be an unqualified success, it is at least ingenious and progressive, especially in the layout of deck space and equipment. This latter includes four seaplanes, with enclosed hangars for two machines, and two catapults, all neatly stowed away without marring the profile of the ship. Finally, if it be true that the radius is 13,000 miles at 15 knots, as compared with the 10,400 miles at 14 knots of the British *Counties*, the American designers must be congratulated on a brilliant technical achievement. Seven further ships of the same generic type are under construction, bringing the total of the "10,000-ton" 8-inch gun ships to fifteen, and giving the United States "parity" in this class of vessel.

It was intended to lay down this year a flotilla leader and ten destroyers, but in deference to economy only five boats have been ordered. The displacement is to be 1500 tons and the cost nearly £1,000,000 per vessel, but no other details are to hand. With the completion of the submarines *Narwhal* and *Nautilus*, of 2760 tons, it was decided to make a drastic cut in dimen-

sions, and the *Dolphin*, *Cachalot*, and *Cuttlefish*, now on the stocks, displace only 1560 tons. The armament is to be one 4-inch gun and six torpedo tubes, in place of the two 6-inch guns and the six tubes carried in the larger boats. On the other hand, the surface speed is to be increased from 17 knots to 18.

The aircraft carrier *Ranger*, of 13,800 tons, laid down at Newport News in 1930, is due to be launched this year. Her speed is reported to be 33 knots, her armament eight 5-inch guns, and her capacity seventy-six aeroplanes. A sister ship may be laid down in the present year.

JAPAN.

The *Chokai*, launched on April 5, was the last of the Japanese 10,000-ton cruisers to go afloat, and is the eighth of her type. All exhibit the same remarkable profile, with an undulating deck line, sharply raking, trunked-in funnels, massive bridge-work, heavy director towers, and the pyramidal disposition of turrets in the forepart of the ship. Like their first American contemporaries, they mount ten 8-inch guns, but in the Japanese ships the twin-mounting system is retained, and both vertical and deck armor protection is stated to be very substantial. Finally, the machinery generates 130,000 S.H.P. for a speed of 33 knots. How the designers have contrived to compress all these qualities into 10,000 tons is a secret known only to themselves. If the details are reliable, these cruisers are, indeed, the "wonder ships" of the day, but it is possible that some of their features are exaggerated. Besides the eight 10,000-ton ships, Japan has completed four cruisers of 7100 tons, armed with six 8-inch guns apiece. Her quota of 8-inch gun ships is therefore filled, and in common with this country she is debarred by the London Treaty from building any further ships with this armament until 1936. It is not the least anomalous feature of the London covenant that it should preclude the British Empire and Japan from building a type of ship which continues to be actively developed by the United States, France and Italy. As our last big 8-inch-gun cruiser was begun in September, 1927, it follows that she will be outclassed by foreign ships which have either just been started or will not be laid down for another year or so. It is a pity that this obvious fact was overlooked by the British delegates who helped to draft the London Treaty.

After the *Chokai*, the largest Japanese ship of war to be launched during the past year was the aircraft carrier *Ryujo*, of 7600 tons. Her principal characteristics are:—Length, 548 feet; breadth, 60 feet 6 inches; turbine machinery of 40,000 S.H.P., for 25 knots; armament, twelve 5.1-inch anti-aircraft guns. This vessel, which was laid down at Yokohama in January, 1930, appears to be a reduced version of the *Hosho*, and, like her, will be fitted with gyro-stabilizers. She is the fourth aircraft carrier to be built for the Japanese Navy. A number of destroyers were launched during the year. They belong to the *Fubuki* class, twenty-four in all, the nameship of which was launched in 1927, and they are the most powerful destroyers, as distinct from leaders, which have yet been built. The dimensions are:—Length, 367 feet; breadth, 34 feet; draught, 10¾ feet; displacement, 1700 tons; machinery, Parsons geared turbines and Kampon boilers, developing 50,000 S.H.P., the designed speed being 35 knots. The armament comprises six 5.1-inch guns, two A.A. guns, and nine 21-inch torpedo tubes on triple mountings. The 5.1-inch guns are mounted in pairs, each mounting enclosed in an armored gas-proof shield. This is a decided novelty in destroyer armament. The torpedo broadside is exceptionally heavy; in fact, the whole armament is the heaviest ever mounted in destroyers. The Japanese principle of en-

dowing every naval unit with the maximum fighting power compatible with its tonnage finds emphatic expression in these remarkable craft. Submarine construction appears to have been slowed down, only five new boats being reported on the stocks at the time of writing. They are vessels of 1638 tons and a surface speed of 19 knots. There are now seventy-two Japanese submarines completed or building, twenty-three of which are capable of 19 knots.

FRANCE.

A battle-cruiser of 26,000 tons is among the ships projected for commencement this year. Originally planned to displace 23,330 tons and to carry 12-inch guns, the vessel has been re-designed to mount eight 13.4-inch guns. The construction of such a unit has long been urged in French naval circles to counteract the German "pocket battleship" *Deutschland*, of 10,000 tons and six 11-inch guns, but it must be admitted that the French "reply" is rather overwhelming. The ship, provisionally named *Dunquerque*, will probably be laid down at Brest.

In December the 10,000-ton cruiser *Algérie* was launched at Brest, almost exactly twelve months after the laying of her keel. The seventh vessel of this type to go afloat for the French Navy, she differs from her predecessors in having lower speed—31 knots, as against 33–34—and more substantial protection, a 6-inch belt on the water line taking the place of the flimsy plating of earlier ships. Moreover, the turrets and hoists are to be strongly armored instead of being merely splinter-proof, as in previous designs. The main armament of eight 8-inch guns is unchanged, but the A.A battery is to consist of twelve 4-inch guns. The *Algérie*, like the Italian *Zara*, appears to be a genuine armored cruiser, and should therefore be a better fighting unit than the majority of 10,000-ton cruisers, to which the term "tin-clad" has been not inaptly applied. The *Foch*, a 10,000-tonner launched in 1929, was completed during the year under review. It is doubtful whether France will build any further ships of this tonnage, for the six cruisers authorized in the programs of 1930 and 1931 are 7500 tons. The first two, *La Galissonnière* and *Jean de Vienne*, have yet to be laid down, the design having undergone modification since its inception. Six-inch guns will be mounted, either eight in twin or nine in triple turrets, and a speed of at least 35 knots is foreshadowed.

A new minelaying cruiser, *Emile Bertin*, is to be launched this year at St. Nazaire. She is an improved version of the *Pluton*, completed last April, which is a ship of 4850 tons and 30 knots speed, carrying four 5.5-inch guns and 250 mines. The *Bertin* will exceed 6000 tons, her speed will be 30 knots, and she is to mount nine 6-inch guns in triple turrets. This will be the first appearance in the French Navy of the 6-inch triple mounting, which has been adopted in all the German post-war cruisers, though its superiority over the twin mounting as regards volume of fire is yet to be demonstrated. Another "special type" cruiser completed during the year is the *Jeanne d'Arc*, designed primarily as a training ship for naval cadets. She is 525 feet long by 57½ feet beam, has turbine engines for 25½ knots, and a main armament of eight 6.1-inch guns. From 1922 to 1931, inclusive, France has built and authorized nineteen cruising ships, as against twenty-three provided for the British Empire.

Four large flotilla leaders were launched in the past year, and twelve others are now on the stocks. They are ships of 2441 to 2500 tons, with a designed speed of 36 knots, and an armament of five 5.5-inch guns and six to eight 21.7-inch torpedo tubes. Thirty of these leaders have now been built or laid down, and a further unit—2600 tons and 42 knots—is to be started this year. The type is peculiar to the French Navy, and although

very impressive on paper its functions appear to be obscure. Vessels of this size and cost are quite unsuitable for ordinary torpedo work, and at the same time are unnecessarily large for dealing with hostile destroyers. Theoretically they are powerful enough to engage light cruisers, but in practice the latter possess a marked advantage by reason of their superior steadiness as gun platforms. The French leaders, it is true, have given some astonishing steaming performances, speeds up to 42.78 knots having been attained on trial, but the tactical value of speed is still a subject of controversy. The inclusion in this year's program of a solitary destroyer, experimental in design, may presage a reversion to smaller dimensions in French torpedo craft construction.

Among the forty-two submarines in hand at the close of the year are units of the *Redoutable* class, which with slight modifications has been steadily developed since 1924. The latest boats are of 1384 tons on the surface and 2080 tons submerged. All are equipped with two sets of Sulzer-Diesel engines with a combined output of 6000 H.P., giving a surface speed of 18 knots, usually exceeded by 1 knot on trial. Electromotors of 2000 H.P. produce a speed of 10 knots below the surface. The hull is 302 feet by 30½ feet, and is very strongly built. These vessels can cruise at sea for thirty days. They are armed with a 3.9-inch gun and eleven 21.7-inch torpedo tubes, including two pairs of revolving carriages. Every boat is required on completion to make an independent voyage of several weeks' duration. The type has given evidence of seaworthiness and engine-room reliability under all conditions, and is unquestionably one of the best submarine designs extant. The minelaying submarine *Rubis*, the sixth of her class, was launched on September 30. The surface displacement is 670 tons, and the speed 12 knots. Armament: One 3-inch gun, four torpedo tubes, and thirty-two mines carried in separate chutes. A large number of second-class submarines are under construction, thirty-one boats of these types having been laid down since 1924. The latest displace 571 tons, have a speed of 14 knots, and are armed with a 3-inch A.A. gun and eight tubes. The endurance at economical speed is 3000 miles. Eight convoy sloops have been authorized for building, but the design does not appear to have been settled. The first quartet of *Colonial* sloops, *Bougainville* class, was launched during the year. They are Diesel-driven ships of 1968 tons and 15½ knots speed, with a radius of 9000 miles at 10 knots. Each mounts three 5.5-inch guns and four light A.A. pieces, is fitted for laying mines, of which fifty are carried, and is able to carry a seaplane. They are useful and economical ships which compare more than favorably with the British sloops of the *Bridgewater* series. The aircraft tender *Commandant Teste*, of 10,000 tons and 20½ knots speed, was commissioned during the year. She carries seaplanes, and has four catapults, but there is no flying-on deck, and the utility of this ship does not appear to be very great.

During the year trials were continued with the French submarine cruiser *Surcouf*. This vessel, which was laid down at Cherbourg in 1927, has a surface speed of 18 knots and 10 knots submerged, and is the largest submarine in the world. She is armed with two 8-inch guns fixed on a turret which apparently cannot be trained, so that in order to bring the guns on to their target it would appear to be necessary to maneuver the whole ship. The *Surcouf* has a radius of action of 10,000 miles at 10 knots.

ITALY.

The largest ship launched during the year was the 10,000-ton cruiser *Pola*, the sixth of her class. She has better protection than the earlier ships and the speed has been reduced to 32 knots, but the armament of eight 8-inch and

sixteen 3.9-inch guns remains unaltered. A seventh vessel, *Bolzano*, is to go afloat this year. In her case the speed has again been raised to 35½ knots at the expense of protection. Widespread interest has been evoked by the trials of the *Condottieri* cruisers, of 5250 to 5550 tons. Four of these eight ships have been completed, and in each case the designed speed of 37 knots was considerably bettered during trials. The *Alberico di Barbiano* reached 42.048 knots, the *Alberto di Giussano* made 40.7 knots, and the other two came within a fraction of 40 knots. Still more remarkable was the eight hours' endurance test of the *Barbiano* at a mean speed of 39.74 knots. This trial was made with the ship displacing 5607 tons, the full armament being on board. Besides bringing a powerful reinforcement to the Italian Navy, these cruisers reflect exceptional credit on those who have designed, built, and engined them. Another steaming record is claimed on behalf of the large destroyer *Alvise Cadamosto*, of 1654 tons and a designed speed of 38 knots. The machinery has a nominal output of 53,500, but during a recent trial run 71,000 H.P. was obtained and the speed of the ship over the measured mile was 44 knots. Twelve smaller destroyers, from 1225 to 1472 tons and 38 knots, are in various stages of construction. In these craft all the uptakes are led into a single funnel and super-structure has been cut down to the minimum. The twenty-two submarines authorized in 1930 were all begun during the past year. Among them are four of the *Balilla* class, which has proved very successful. The surface displacement is 1390 tons, the speed 18½ knots, and the armament comprises a 3.9-inch gun, six torpedo tubes, and sixteen mines with laying gear. They are specially built for deep diving, and the *Balilla* herself reached a depth of 55 fathoms during her trials. The new coastal type, of which nineteen are building, is of 636 tons with a speed of 14 knots and the same armament as the *Balilla*, except that no mines are carried.

Mention may be made here of the remarkable pair of Argentine cruisers, *Almirante Brown* and *Veinticinco de Mayo*, which were built by the Italian firms of Odero and Orlando, respectively. Displacing only 6800 tons, they have a speed of 32 knots and mount the formidable armament of six 7.5-inch guns in twin turrets, twelve 3.9-inch A.A. guns, and six torpedo tubes. The single large funnel, splayed tripod foremast, and clipper bow give these ships a very distinctive appearance.

GERMANY.

Interest in that much-advertised warship the *Deutschland* was revived by her launch at the Deutsche Werke, Kiel, on May 31, 1931. She had been thirty-one months on the stocks, and it would be interesting to know whether this slow rate of construction was due to complexity of design, the extensive use of electric welding, or financial reasons. Delivery of this "pocket battleship" is expected to be made about four years after the laying of her keel. It may be convenient to restate the principal features of the design. The ship has a length of 609 feet 3 inches overall, with a breadth of 67 feet 6 inches, and draws 21 feet 8 inches, the "standard" displacement being 10,000 tons, this figure being the maximum permitted by the Versailles Treaty. Everything possible has been done to save weight. Only steel of the highest quality is used in the construction, and wherever feasible the electric torch has replaced the riveting hammer. As a result the hull is claimed to be over 500 tons lighter than it would have been if constructed on conventional lines. An armor belt of unknown strength has been worked into the hull amidships, and there are double protective decks over the more vulnerable spaces. Extensive subdivision has been employed as a defense against under-water

attack. As the forecastle deck is carried well aft the ship has a fairly high freeboard, and in spite of the heavy deck load which she carries, the model tank experiments give promise of stability and seaworthiness. For her tonnage the *Deutschland* will be the most heavily armed vessel in the world. She will mount six 11-inch 50-caliber guns in triple turrets fore and aft; eight 5.9-inch Q.F. mounted singly on deck behind shields; four 3.4-inch A.A. on twin mounts, and two sets of triple 19.7-inch torpedo tubes, one at each side of the quarterdeck. There is a spacious bridge surmounted by an armored gun control tower with a 30-foot base range-finder, a heavy steel foremast carrying the director top, and one large funnel.

The *Deutschland* will be the first armored ship of considerable tonnage to be propelled entirely by internal combustion engines, and full details of her machinery are now available. It comprises eight compressorless, double-acting, two-stroke, M.A.N. Diesel engines, each set of four engines driving a single propeller shaft through a Vulkan hydraulic gear. Each of the nine working cylinders of an engine unit has a diameter of 420 millimeters and a piston stroke of 580 millimeters. At full power, 450 R.P.M., the plant develops 56,800, and when allowance has been made for loss of power through the hydraulic coupling and the gears it is estimated that 54,000 H.P. will be transmitted to the propellers. It should be added that the Vulkan gear reduces the speed of the shafts to 250 R.P.M. The entire installation is said to be unique in the power developed proportionate to its weight. At the same time the original claim that a unit of horsepower was generated for every 17½ pounds of weight is now shown to be misleading. The gross weight works out at 45½ pounds per unit of power, and while this achievement represents a notable advance in the fining-down of internal combustion engine weight, it is inferior to the ratio of weight to power exhibited by the latest steam turbine installations in naval practice. From private but trustworthy information I gather that the service figure for the *Deutschland* will be 51 pounds per horsepower. Against this figure the British Navy has developed a turbine plant for light cruisers which produces a unit of power for little over 40 pounds of weight, while in the latest destroyer turbine installations there is a saving of at least 15 pounds as compared with the *Deutschland's* figure. This fact in no way detracts from the technical achievement represented by the engines of the German ship, but it does suggest that the Diesel system is still at a serious disadvantage in comparison with the steam turbine as a prime mover for high-speed fighting ships. Moreover, as the *Deutschland* is not yet completed, the power and efficiency of her propelling plant are still to some extent speculative. She constitutes a bold and praiseworthy experiment—but still an experiment, which may or may not fulfil the somewhat ambitious promises of her authors.

Equally significant from the naval point of view are the cruisers which have been recently completed for the German Navy. In four of these ships the steam turbines which form the main propelling machinery are supplemented by an auxiliary Diesel plant for cruising purposes. The sister ships *Königsberg, Karlsruhe,* and *Köln,* have the following characteristics:—Length overall, 570 feet; breadth, 49 feet 10½ inches; draught, 17 feet 9 inches; displacement, 6000 tons. Electric welding was largely employed in building the hulls. Each vessel is armed with nine 5.9-inch guns in triple turrets, one turret being forward and the other two aft, where they are echeloned—presumably to facilitate the layout of magazines. There are also four 3.4-inch A.A. and twelve torpedo tubes on deck in triple mountings. The steam turbines develop 65,000 S.H.P., the designed speed being 32 knots. No provision was made in the original design for an auxiliary Diesel

drive, which was literally an afterthought. It was found that the necessary space was available, and that the additional weight could be accommodated without exceeding the 6000 tons of total displacement. The plant installed in the *Königsberg* and her two sisters comprises two Diesel motors, each of 1000 B.H.P., working through Vulkan reduction gear. For economical reasons the actual output is restricted to 900, so that the combined power of the auxiliary drive is 1800 B.H.P. Intended solely to propel the ship at moderate cruising speed, it adds nothing to her speed when the turbines are running at full power. Its great merit is that it confers upon these small ships a cruising endurance of 18,000 miles, enabling them to cover long distances at a very moderate consumption of fuel. This extraordinary endurance is, of course, a valuable strategic asset, since the vessels are almost independent of shore bases. The ten working cylinders in each motor are 260 millimeters diameter and have a piston stroke of 330 millimeters, the net engine weight being 12.3 pounds per unit of horsepower—a remarkably low ratio. It is ascribed in part to the use of aluminium castings for the engine frames and pistons and in part to meticulous care in design.

In the cruiser *Leipzig*, which was commissioned on October 8, 1931, the system of auxiliary drive has been still further developed. This ship has much the same dimensions and the same armament as her predecessors of the *K* class, and the steam turbine machinery is but little altered. Instead of two shafts, however, there are three, the center one being driven by the Diesel plant. The latter consists of four seven-cylinder motors generating 3100 B.H.P. each, the total output for service purposes being calculated at 12,000. As this power is sufficient to drive the ship at 18 knots, the Diesel plant, in the case of the *Leipzig*, is something more than a mere "auxiliary." The difficulty of synchronizing the revolutions of the center, or Diesel-driven, shaft with those of the wing shafts operated by the turbines was overcome by fitting that shaft with a special propeller with adjustable blades. When the ship is running on her turbines alone the blades of the middle propeller offer only slight resistance. Conversely, when the Diesel motors are working and the turbines cut out, the wing propellers are turned by small electromotors, fed by current from a dynamo driven by the center shaft through a simple form of gearing. By this means the wing shafts revolve at a speed so regulated as substantially to reduce the loss of power from resistance that would result if they were left to turn freely. Much ingenuity, it will be gathered, has been displayed in the solution of the various problems to which this combination of turbine and Diesel propulsion gave rise.

The small cruiser *Bremse*, launched at Wilhelmshaven on January 24, deserves special notice as being the first cruising warship to have all-Diesel drive. Officially rated as a gunnery tender, she is 318 feet long, 31 feet 3 inches in breadth, and displaces 1225 tons. The armament is four 4.1-inch guns. Propulsion is effected by eight motors, four to a shaft, working through Vulkan reduction gearing. They are of the same design as the *Leipzig's* motors, except that an additional cylinder has been added—eight instead of seven—and the total output of the plant is 26,000 B.H.P., estimated to give a speed of 27 knots. Here, then, is a Diesel-driven cruiser of really high speed, which also possesses the advantage of a radius of action from twice to three times as great as that of a steamer of equivalent tonnage and power. The *Bremse* worthily completes a series of German post-war naval craft which deserve full recognition as examples of technical skill and, above all, of that progressive spirit in naval design which has not been conspicuous of late in certain other countries. German constructors have not been content to abide by the prevailing conventional standards which elsewhere have produced a large number of warships of painfully mediocre

design. The commercial value to Germany of her pioneer work in developing the internal combustion engine, exemplified in her latest naval vessels, is too obvious to need emphasis.

OTHER NAVIES.

That British shipbuilders are second to none in originality of conception and fine workmanship is proved by the ships which they have constructed recently for foreign Governments. The six large destroyers built by Thornycrofts in 1928 for the Chilean Navy, and the three Argentine flotilla leaders launched by J. Samuel White and Co., in 1928-29, are among the finest torpedo craft produced since the war. Equally striking has been the success of Yarrow and Co. in securing contracts for the design of all the new destroyers of the Netherlands Navy. The plans submitted by this firm were chosen out of many prepared by Continental builders, and the eight vessels which have been built to the British firm's drawings have given full satisfaction. The same firm has since designed a Netherlands flotilla leader, the details of which have not yet appeared, and is also building for the Jugo-Slav Navy a leader of exceptional size and power. This vessel, to be known as the *Dubrovnik*, was launched at Scotstoun on October 11, and is 371.5 feet in length, with a beam of 35 feet, and her displacement is 1880 tons. She is driven by Parsons turbines with single-reduction gearing, taking steam from three Yarrow boilers, with a designed performance of 42,000 S.H.P., the contract speed being 37 knots. The armament—four 5.5-inch guns, one 3.3-inch A.A. gun, and six 21-inch tubes—is of unusual power for a vessel of this type.

British industry has also secured a share of the Portuguese naval contracts which were placed last year. Hawthorn, Leslie and Co. are building two small turbine cruisers of 1174 tons, while Yarrows have in hand two destroyers, of 1383 tons and 36 knots.

Two coast defense ships of a novel type, built for the Finnish Navy, were launched from the Crichton-Vulcan yard at Abo—the *Vainamoinen* in December, 1930, the *Ilmarinen* on July 9 last. In effect they are sea-going monitors. The displacement is about 4000 tons. Propulsion is on the Diesel-electric system, the contract speed being 16 knots. There is 2¼-inch armor on the sides, reinforced by a protective deck and steel transverse bulkheads. The armament is formidable, consisting as it does of four 10-inch guns, paired in turrets at bow and stern, and eight 4.7-inch Q.F. It is understood that the guns, which are of a new model, have been supplied by the Bofors works in Sweden. The ships draw comparatively little water, they are designed for rapid maneuvering, and in certain conditions they might prove dangerous to any capital ships likely to be encountered in the Baltic. Finland has also built four submarines.

A fact of some significance is the large and growing number of submarines now possessed by the minor maritime States. These craft are to be found in the navies of Finland, Turkey, Jugo-Slavia, Rumania, Poland, Peru and Latvia. There are upwards of forty in the three Scandinavian fleets, and Holland has over thirty boats. Between them the secondary naval Powers possess no fewer than 148 submarines, the great majority being of post-war construction. This figure should give pause to those who treat the future of submarine warfare as if it were a question that concerned the great Powers alone. The almost universal popularity of the submarine suggests that proposals for its abolition, which it is reported are to be raised at the forthcoming Disarmament Conference, are unlikely to emerge from the stage of purely academic discussion.—"The Engineer," Jan. 1 and 8, 1932.

SHIPS UNDER CONSTRUCTION AT THE END OF 1931.

The Quarterly Returns of Lloyd's Register of Shipping.

GREAT BRITAIN AND IRELAND.

The returns compiled by Lloyd's Register of Shipping (issued on the 13th January), which take into account only vessels of 100 tons gross and upwards the construction of which had actually been begun, show that there were 98 merchant ships of 400,505 tons gross (309,970 steam, 89,855 motor, and 680 sailing ships or barges) being built in Great Britain and Ireland at the close of the quarter ended 31st December, 1931. The shipping then under construction was 16,880 tons less than that on hand at the end of the previous quarter, and 508,397 tons less than that at the 31st December, 1930. Moreover, the figure for December, 1931, was the lowest recorded since September, 1887, and included 19 vessels of 153,680 tons on which work had been suspended. These 19 ships presumably included the Cunard liner of about 73,000 tons gross on the stocks at Clydebank.

Of the total tonnage under construction in Great Britain and Ireland at the 31st December, 1931, there were being built for owners in the British Isles 57 ships of 284,390 tons gross, and 26 ships of 68,720 tons for owners abroad. The latter total included 37,235 tons for the British Dominions, 800 tons for Belgium. 16,470 tons for France, 3456 tons for Mexico, 8309 tons for Norway, and 2460 tons for Jugoslavia. There were also being built 15 ships of 47,395 tons for sale or country not stated.

Table I.—Merchant Tonnage under Construction in Great Britain and Ireland.

Districts.	Dec., 1931.		Sept., 1931.		Dec., 1930.	
	No.	Tons.	No.	Tons.	No.	Tons.
Aberdeen	3	520	1	120	11	3,554
Barrow, Maryport and Workington	3	45,783	2	22,340	4	48,340
Belfast	4	56,130	6	75,957	11	135,800
Bristol	1	150	1	150	1	155
Clyde :—						
Glasgow	19	109,319	22	137,160	41	261,847
Greenock	16	78,860	15	61,420	20	115,650
Dublin	—	—	1	600	—	—
Dundee	1	720	—	—	4	32,376
Hartlepool	6	26,350	1	5,400	—	—
Hull	5	1,437	4	1,270	15	5,958
Leith	5	11,251	4	7,190	5	10,160
Liverpool	4	6,220	2	5,000	9	23,270
Middlesbrough, Stockton and Whitby	12	10,986	11	9,055	10	43,930
Newcastle	9	44,304	10	80,318	29	193,384
Southampton	2	415	—	—	5	1,810
Sunderland	2	7,020	5	10,920	11	31,518
Elsewhere	6	1,040	3	485	5	1,150
Total	98	400,505	88	417,385	181	908,902

Table I shows how the number and gross tonnage of merchant ships being built at the 31st December, 1931, were divided among the principal shipbuilding districts, as compared with the figures at the 30th September, 1931, and 31st December, 1930.

Ships launched in Great Britain and Ireland during the quarter under review numbered 24, of 70,707 tons gross, as compared with 23 vessels of 80,340 tons during the previous quarter. The 24 ships sent off the stocks in the December quarter included 20 steamships of 28,388 tons and four motorships of 42,319 tons, no sailing ships or barges having been launched during the period.

Vessels begun during the quarter ended 31st December, 1931, numbered 37, of 104,784 tons gross, as compared with 25 vessels of 38,675 tons during the previous quarter. The 37 ships laid down in the quarter just ended included 30 steamships of 103,349 tons, four motorships of 1015 tons, and three sailing ships or barges of 420 tons.

TONNAGE UNDER CONSTRUCTION ABROAD.

The total merchant tonnage being build abroad at the 31st December, 1931, consisted of 183 ships of 1,003,290 tons gross, including 72 steel steamships of 461,431 tons, one wood steamship of 150 tons, 103 steel motorships of 538,728 tons, two wood motorships of 1500 tons, and five steel sailing ships or barges of 1481 tons. The tonnage under construction abroad was 110,445 tons less than that in hand at the end of September, 1931.

Table II shows how the number and gross tonnage of merchant ships under construction abroad at the 31st December, 1931, were divided among the various countries, as compared with the figures at the 30th September, 1931, and the 31st December, 1930.

TABLE II.—MERCHANT TONNAGE UNDER CONSTRUCTION ABROAD.

Countries.	Dec., 1931.		Sept., 1931.		Dec., 1930.	
	No.	Tons.	No.	Tons.	No.	Tons.
British Dominions	4	3,101	7	3,551	14	12,184
Belgium	3	1,810	3	1,810	4	6,825
Brazil	1	125	1	125	1	125
China	—	—	1	850	2	730
Danzig	2	1,920	2	4,840	14	12,070
Denmark	12	51,800	12	69,160	23	107,660
France	17	164,440	12	169,720	19	174,215
Germany	17	103,981	20	113,468	58	218,215
Holland	26	67,866	31	95,216	50	160,078
Italy	15	178,287	20	159,147	24	179,677
Japan	17	53,280	11	32,620	16	86,060
Lithuania	2	950	2	950	2	950
Norway	16	15,785	23	25,440	18	21,260
Portugal	2	1,337	1	355	—	—
Spain	10	55,241	10	53,889	12	59,355
Sweden	18	95,380	20	121,080	29	145,750
United States	21	207,837	25	261,364	35	232,030
Uruguay	1	150	1	150	—	—
Total	183	1,003,290	202	1,113,735	321	1,417,184

Ships launched abroad during the quarter ended 31st December, 1931, numbered 66, of 230,971 tons gross, including 22 steamships of 131,859 tons and 44 motorships of 99,112 tons, no sailing ships or barges having been launched during the period. Vessels of 73,534 tons were sent off the stocks in the United States, 68,646 tons in Italy, 21,500 tons in Sweden, 18,176 tons in Holland, 14,850 tons in Denmark, and 13,396 tons in Japan. The ships launched abroad showed a decrease of 75,297 tons as compared with the output of the previous quarter.

Ships begun abroad during the quarter ended 31st December, 1931, numbered 47, of 120,698 tons gross, including 16 steamships of 39,454 tons and 31 motorships of 81,244 tons. No sailing ships or barges were begun abroad during the period under review. Vessels of 38,640 tons were laid down in Japan, 32,300 tons in Italy, 12,550 tons in Sweden, and 10,400 tons in the United States. The tonnage begun abroad showed a decrease of 13,987 tons as compared with that laid down in the previous quarter.

OIL-CARRYING SHIPS.

The returns show that at the 31st December, 1931, there were 45 vessels (41 motorships and four steamships), each of 1000 tons and above, with a total gross tonnage of 351,320, under construction throughout the world for the carriage of oil in bulk, as detailed in Table III. This is a decrease on the corresponding figures three months earlier, which were 64 ships of 505,258 tons. The tanker tonnage being built at the 31st December represented 25 per cent of the total steam and motor tonnage under construction.

TABLE III.—OIL-TANK SHIPS OF 1,000 TONS GROSS AND ABOVE UNDER CONSTRUCTION THROUGHOUT THE WORLD.

Countries where being Built.	Steam.		Motor.		Total.	
	No.	Tons.	No.	Tons.	No.	Tons.
Great Britain and Ireland	4	18,531	6	46,910	10	65,441
Denmark	—	—	3	24,535	3	24,535
Germany	—	—	6	72,400	6	72,400
Holland	—	—	3	24,550	3	24,550
Italy	—	—	3	33,900	3	33,900
Norway	—	—	1	6,360	1	6,360
Spain	—	—	6	39,200	6	39,200
Sweden	—	—	12	83,400	12	83,400
United States	—	—	1	1,534	1	1,534
Total	4	18,531	41	332,789	45	351,320

MOTOR-DRIVEN SHIPS.

The vessels being built throughout the world at the end of December, 1931, which were to be propelled by internal-combustion engines, aggregated 630,083 tons gross, as compared with 776,431 tons at the 30th September, 1931. The motorships under construction are dealt with elsewhere in the present issue.—"The Shipbuilder and Marine Engine-Builder," February, 1932.

OUTPUT IN SHIPYARDS OF THE UNITED STATES DURING 1931.

Shipbuilding returns for 1931, as compiled by "Marine Engineering and Shipping Age," indicate that, in spite of the current depression, the output of American shipyards was greater than either of the pre-depression years, 1928 and 1929. A comparison of the present returns with those of 1930, however, shows a decrease in the tonnage of vessels built and also under construction at the end of the year. This reduction is due to the fact that the first requirements under the Jones-White Act of 1928 have been fulfilled.

Mail contract requirements under the terms of this act call for the construction of six seagoing passenger and cargo ships for delivery in 1933. These must be contracted for in 1932. With the volume of ship construction under way at the end of 1931, it is expected that the tonnage output for 1932 will be practically normal.

Under the terms of the Jones-White Act of 1928, in addition to those vessels already built or under construction, 32 new ships must be built and 35 vessels must be reconstructed. Of the new vessels required six must be in service in 1933, three in 1934, seven in 1935, one in 1936, two in 1937, three in 1938, one in 1939, one in 1940, one in 1941 and seven during the term of contract. Of the vessels which required reconditioning twelve must be completed in 1932, one in 1934, one in 1935, one in 1937, one in 1938 and nineteen during the term of the contract.

During the year contracts were placed for eight passenger and cargo ships, two freight car carriers, one tanker, one scout cruiser, one submarine, two destroyers and a large number of miscellaneous vessels of under 1000 gross tons each. Vessels delivered during 1931 include eight passenger and cargo ships, five tankers, two yachts, one scout cruiser and a Coast Guard cutter.

During 1931, 291 merchant vessels of all types, totaling 301,810 gross tons and 5 naval vessels totaling 32,200 displacement tons were delivered from 58 shipyards. Thirty-eight of the yards reported 99 merchant vessels of 244,190 gross tons and 16 naval vessels of 93,800 displacement tons under construction at the end of 1931. The output in 1931 represents a decrease of 11.2 per cent as compared with that of 1930, while the tonnage under construction at the end of the year shows a decrease of 25 per cent as compared with that on order at the end of 1930. The output in 1931, however, represents an increase of 6.3 per cent over that of 1929 and an increase of 30.8 per cent over that of 1928. Orders on hand showed a decrease of 30.1 per cent as compared with that of 1929 but an increase of 134 per cent over that of 1928. Sixty-nine yards were active during the year as compared with 76 in 1930, 63 in 1929, 44 in 1928 and 51 in 1927. The 38 yards reporting orders compared with 41 in 1930, 48 in 1929, 33 in 1928 and 43 in 1927.

Of the vessels delivered in 1931, 43.6 per cent were propelled by steam machinery; 17.8 per cent consisted of motorships and the remaining 38.6 per cent were barges and other non-propelled craft. Of the vessels under construction 86.7 per cent are steamships; 2.1 per cent are motorships; while barges and non-propelled craft represent 11.2 per cent.

Of the tonnage built in 1931, Atlantic coast and Gulf shipyards constructed 65.1 per cent; Great Lakes yards built 6.3 per cent; western river plants supplied 26.8 per cent; while the west coast yards are credited with 1.8 per cent. The Atlantic coast and Gulf yards are handling 87.4 per cent of the tonnage now under construction; the Great Lakes yards 2.3 per cent; the western river yards 9.7 per cent and the west coast yards 0.6 per cent.

Table 1—Merchant Vessels Completed in 1931

	Number of Vessels	Total Gross Tonnage	Total Horse-power
Steamships	31	131,779	153,470
Motorships	70	53,287	50,990
Non-propelled	190	116,744	.....
Total	291	301,810	204,460

Table 2—Merchant Vessels Now Under Construction

	Number of Vessels	Total Gross Tonnage	Total Horse-power
Steamships	28	211,744	259,920
Motorships	20	5,117	13,295
Non-propelled	51	27,329	.....
Total	99	244,190	273,215

Table 3—Steamships Built in 1931

Name of Company	Number of Vessels	Total Gross Tonnage	Total Horse-power
Bethlehem Shipbuilding Corporation, Ltd. (Fore River)	4	32,060	33,520
Dravo Contracting Company	1	1,800	1,800
Great Lakes Engineering Works	3	3,825	3,900
Great Lakes Towing Company	2	188	1,500
Jones & Laughlin Steel Corporation	1	190	750
George Lawley & Son Corporation	2	522	1,900
Manitowoc Shipbuilding Company	2	3,175	3,550
Marietta Manufacturing Company	1	700	1,100
Newport News Shipbuilding & Dry Dock Company	4	56,300	72,850
New York Shipbuilding Company	3	28,080	24,000
Pusey & Jones Corporation	3	707	2,100
Todd Shipyards Corporation	2	1,187	1,300
United Dry Docks, Inc.	1	2,045	4,000
Chas. Ward Engineering Works	2	1,000	1,200
Total	31	131,779	153,470

Table 4—Motorships Built in 1931

Name of Company	Number of Vessels	Total Gross Tonnage	Total Horse-power
Bath Iron Works Corporation	9	4,604	10,140
Bethlehem Shipbuilding Corporation, Ltd. (Union Plant)	1	333	1,500
(Baltimore Plant)	2	1,372	700
Ira S. Bushey & Sons, Inc.	1	300	240
Calumet Shipyard & Dry Dock Co.	1	900	250
Charleston Dry Dock & Machine Company	1	100	125
Craig Shipbuilding Company	2	1,009	3,000
Defoe Boat & Motor Works	4	460	2,500
Dubuque Boat & Boiler Works	1	1,540	2,200
Electric Boat Company	1	101	150
General Engineering & Drydock Company	1	45	125
Hampton Roads Shipbuilding Corporation	1	68	225
Harbor Boat Building Company	2	500	700
Robert Jacob, Inc.	1	13	130
Larsen Shipyard	1	85	250
George Lawley & Sons Corporation	8	172	625
Marietta Manufacturing Company	4	900	1,420
Marine Construction Company	4	40	320
Marine Iron & Shipbuilding Company	4	384	940
Merrill-Stevens Drydock & Repair Company	1	323	250
Midland Barge Company	1	684	560
Moore Dry Dock Company	2	797	990
Pusey & Jones Corporation	3	1,582	2,625
Rice Brothers Corporation	1	108	600
Southern Shipyard Corporation	6	150	6,600
Sun Shipbuilding & Dry Dock Company	5	36,448	11,695
Todd Shipyards Corporation	1	269	2,130
Total	70	53,287	50,990

Table 5—Non-Propelled Vessels Built in 1931

Name of Company	Number of Vessels	Total Gross Tonnage
Alabama Dry Dock & Shipbuilding Company	12	13,282
American Bridge Company	10	11,750
American Shipbuilding Company	1	1,250
Bethlehem Shipbuilding Corporation, Ltd. (Union Plant)	3	2,256
(Baltimore Plant)	6	2,068
Brewer Dry Dock Company	2	600
Ira S. Bushey & Sons, Inc.	1	350
Dravo Contracting Company	72	33,365
Federal Shipbuilding & Dry Dock Company	1	743
Hegewald Company	4	.....
Howard Shipyards & Dock Company	1	820
Ingalls Iron Works	5	9,000
Knowles Iron Works, Inc.	1	300
R. Lenahan Company	3	.....
Manitowoc Shipbuilding Company	8	3,835
Marietta Manufacturing Company	6	9,600
Marine Construction Company	3	.....
Maryland Dry Dock Company	2	403
McClintic Marshall Company	18	13,015
Midland Barge Company	11	8,040
New Orleans Coal & Bisso Towboat Company	1	120
Norfolk Shipbuilding & Dry Dock Corporation	2	750
L. D. Smith Dock Company	1	2,500
Sun Shipbuilding & Dry Dock Company	7	1,407
United Dry Docks, Inc.	4	350
Welded Shipbuilding, Inc.	1	740
Willamette Iron & Steel Works	4	200
Total	190	116,744

Table 6—Steamships Now Under Construction

Name of Company	Number of Vessels	Total Gross Tonnage	Total Horse-power
Bethlehem Shipbuilding Corporation, Ltd. (Fore River Plant)	5	58,534	75,500
Defoe Boat & Motor Works	1	1,000	1,500
Dravo Contracting Company	2	3,250	2,800
Federal Shipbuilding & Dry Dock Company	4	42,000	50,000
Hampton Roads Shipbuilding Corporation	1	700	1,000
Howard Shipyards & Dock Company	2	1,460	2,500
Marietta Manufacturing Company	2	4,000	3,000
Newport News Shipbuilding & Dry Dock Company	6	37,200	58,800
New York Shipbuilding Company	2	60,000	60,000
Sun Shipbuilding & Dry Dock Company	2	1,600	1,600
United Dry Docks, Inc.	1	2,000	3,220
Total	28	211,744	259,920

Table 7—Motorships Under Construction

Name of Company	Number of Vessels	Total Gross Tonnage	Total Horse-power
Bath Iron Works Corporation	6	2,102	7,800
Berg Shipbuilding Company	1	1,450	1,500
Bethlehem Shipbuilding Corporation, Ltd. (Baltimore Plant)	1	.....	750
Ira S. Bushey & Sons, Inc.	2	160	300
Canulette Shipbuilding Company, Inc.	2	350	800
Cowles-Lenahan Dry Dock, Inc.	1	90	150
M M. Davis & Sons	1	550	600
Howard Shipyards & Dock Company	1	.....	.....
Jakobson & Peterson, Inc.	1	.....	135
George Lawley & Son Corporation	1	85	200
Norfolk Marine Railway Company, Inc.	1	65	360
Leathem D. Smith Dock Company	1	90	300
Spedden Shipbuilding Company	1	175	400
Total	20	5,117	13,295

Table 8—Non-Propelled Vessels Now Under Construction

Name of Company	Number of Vessels	Total Gross Tonnage
Alabama Dry Dock & Shipbuilding Company	4	5,303
Brewer Dry Dock Company	1	400
Canulette Shipbuilding Company, Inc.	10	1,500
Chicago Bridge & Iron Works	3	360
Dravo Contracting Company	3	331
Dubuque Boat & Boiler Works	1	1,000
Great Lakes Engineering Works	4	1,600
R. Lenahan Company	1	.....
Marietta Manufacturing Company	1	150
McClintic-Marshall Company	15	11,025
Midland Barge Company	5	3,960
Nashville Bridge Company	1	1,000
United Dry Docks, Inc.	2	700
Total	51	27,329

Table 9—Naval Vessels Completed in 1931

Name of Company	Number of Vessels	Total Displacement Tonnage	Total Horse-power
Mare Island Navy Yard	1	10,000	107,000
Newport News Shipbuilding & Dry Dock Company	1	10,000	107,000
Portsmouth Navy Yard	2	2,200	4,000
Puget Sound Navy Yard	1	10,000	107,000
Total	5	32,200	325,000

Table 10—Naval Vessels Now Under Construction

Name of Company	Number of Vessels	Total Displacement Tonnage	Total Horse-power
Bath Iron Works Corporation	1	1,500	30,000
Bethlehem Shipbuilding Corporation, Ltd.	2	11,500	137,000
Boston Navy Yard	1	1,500	30,000
Electric Boat Company	1	1,100	2,000
Mare Island Navy Yard	1	10,000	107,000
Newport News Shipbuilding & Dry Dock Company	1	13,000	107,000
New York Shipbuilding Company	2	20,000	214,000
New York Navy Yard	2	11,500	137,000
Philadelphia Navy Yard	1	10,000	107,000
Portsmouth Navy Yard	2	2,200	4,000
Puget Sound Navy Yard	2	11,500	137,000
Total	16	93,800	1,012,000

Outstanding among the vessels completed during the year are the 22,000-ton turbo-electric drive passenger and cargo steamships *President Hoover* and *President Coolidge* built by the Newport News Shipbuilding & Dry Dock Company, Newport News, Va., for the Dollar Steamship Lines, Inc., Ltd., San Francisco, Calif. In addition, this company delivered the 5000-ton passenger and cargo steamer *Florida* to the Peninsula & Occidental Steamship Company, Jacksonville, Fla., and the 7500-ton turbo-electric steamship *Talamanca* to the United Fruit Company, Boston, Mass.

The Fore River Plant of the Bethlehem Shipbuilding Corporation, Ltd., Quincy, Mass., completed the 18,500-ton passenger and cargo steamship *Mariposa* for the Matson Navigation Company, San Francisco, Calif.; the 7100-ton passenger and cargo liner *Borinquen* for the New York & Porto Rico Steamship Company, New York, and the 6100-ton tanker *Harry F. Sinclair* for the Sinclair Navigation Company, New York.

The 9300-ton Export Steamship Company liners *Exochorda, Exeter* and *Excambion* were delivered by the New York Shipbuilding Company, Camden, N. J., for passenger and freight service to the Mediterranean Sea.

The Sun Shipbuilding & Dry Dock Company, Chester, Pa., completed the three 9000-ton Diesel tankers *Northern Sun, Southern Sun* and *Mercury Sun* for the Motor Tankship Corporation, Philadelphia, Pa. This company also delivered the 9100-ton Diesel tanker *Daylight* to the Standard Transportation Company, New York.

The Bath Iron Works Corporation, Bath, Me., completed the 1900-ton turbo-electric yacht *Caroline* for Eldridge R. Johnson and the 1300-ton Diesel yacht *Aras* for Hugh J. Chisholm.

Government-owned vessels delivered in 1931 include the 10,000-ton scout cruiser *Augusta* by the Newport News Shipbuilding & Dry Dock Company, Newport News, Va., and the 2000-ton Coast Guard cutter *Shoshone* by the General Engineering & Dry Dock Company, Oakland, Calif.

Among the outstanding vessels now under construction are the 30,000-ton United States linen *Manhattan* and her sister ship and two 10,000-ton scout cruisers being built by the New York Shipbuilding Company, Camden, N. J.

The Newport News Shipbuilding & Dry Dock Company, Newport News, Va., has under construction the two 7500-ton United Fruit liners *Segovia* and *Chiriqui;* the 13,800-ton airplane carrier *Ranger;* the two 5500-ton Eastern Steamship liners *Saint John* and *Acadia;* and two 5200-ton passenger and cargo steamships for the Colombian Steamship Company, New York.

The Bethlehem Shipbuilding Corporation, Ltd., is building the two 18,500-ton Matson liners, *Monterey* and *Lurline,* the three 7500-ton United Fruit steamships *Antigua, Quirigua* and *Veragua;* the 10,000-ton scout cruiser *Portland;* the 1200-ton Diesel tanker *New York Socony* for the Standard Vacuum Transportation Company, New York; and a 1500-ton destroyer.

Three 9000-ton Diesel tankers for the Motor Tankship Corporation are under construction by the Sun Shipbuilding & Dry Dock Company, Chester, Pa. In addition this company is building two 5000-ton freight car carriers for the Seatrain Lines, Inc., New Orleans.

Four passenger and cargo steamships, each of 10,500 gross tons, are under way at the plant of the Federal Shipbuilding & Dry Dock Company, Kearny, N. J. These vessels, the *Santa Rosa, Santa Paula, Santa Lucia* and *Santa Elena,* are for the Panama Mail Steamship Company, New York.

Miscellaneous vessels under construction include the 2000-ton Coast Guard cutter *Cayuga* being built by United Dry Docks, Inc., New York; a 1100-ton submarine by the Electric Boat Company, Groton, Conn.; and a 1500-ton destroyer by the Bath Iron Works Corp.—"Marine Engineering and Shipping Age," January, 1932.

THE FALL IN MOTORSHIP CONSTRUCTION.

Perhaps the most depressing feature of the shipbuilding returns issued by Lloyd's Register of Shipping on the 13th January, and which are dealt with in detail elsewhere in this number, is the heavy decrease in motorship construction which has taken place in 1931—a decrease which has been proportionately much greater than in the case of steamships. The figures given in Table I demonstrate this fact unmistakably. See pages 246 and 247.

It will be observed that at the end of 1931 the gross tonnage of motorships under construction throughout the world had fallen by 141,468 below that of the steamships being built, whereas at the beginning of that year the motorships exceeded the steamships by 348,675 tons. Abroad the motorships under construction at the end of the year still showed an increase over steamships of 78,647 tons, as compared with 372,415 tons at the beginning of 1931; but in Great Britain and Ireland at the end of the year the motorship tonnage was less by 220,115 than that of the steamships being built, as compared with a difference of only 23,740 tons at the beginning of 1931.

From Table II it will be noted that the indicated horsepower of the marine oil engines under construction throughout the world at the end of 1931 was less than half the horsepower of steam engines being built, *i.e.*, the shaft horsepower of steam turbines and the indicated horsepower of steam reciprocating engines combined.

That motorship construction is not likely in the near future to regain its preponderance over steamship construction is shown by the fact that during the last quarter of 1931 the motorships begun in Great Britain and Ireland only totalled 1015 tons gross and were less than those sent off the stocks in the same period by 41,304 tons; while the steamships commenced exceeded by 74,961 tons those set afloat. Throughout the world the motorships laid down fell short of those launched by 59,172 tons gross, whereas the steamships laid down were only short of those launched by 17,444 tons. It would seem, therefore, that any revival in shipbuilding, measured by tonnage under construction, during the next few months is likely to be due more to the steamship than the motorship.

Table III shows how the number and gross tonnage of motorships being built at the 31st December, 1931, were divided among the home shipbuilding areas; while Table IV gives the corresponding data for all foreign countries. Of the total merchant tonnage (including non-propelled vessels) then under construction in Great Britain and Ireland, 22.44 per cent was to be propelled by internal-combustion engines. Abroad the percentage of motorship tonnage being built was 53.85, and throughout the whole world it was 44.88.

Of the propelling machinery under construction for merchant ships at the 31st December in Great Britain and Ireland, internal-combustion engines represented 15.13 per cent of the total horsepower; abroad the corresponding figure was 37.37 per cent; and throughout the whole world it was 31.82 per cent.—"The Shipbuilder and Marine Engine-Builder," February, 1932.

TABLE I.—GROSS TONNAGE OF STEAMSHIPS AND MOTORSHIPS UNDER CONSTRUCTION IN 1931.

Date.	Great Britain and Ireland.		Abroad.		The World.	
	Steam.	Motor.	Steam.	Motor.	Steam.	Motor.
1st January, 1931	464,016	440,276	520,018	892,433	984,034	1,332,709
31st December 1931	309,970	89,855	461,581	540,228	771,551	630,083
Decrease	154,046	350,421	58,437	352,205	212,483	702,626

TABLE II.—HORSE-POWER OF STEAM AND OIL ENGINES UNDER CONSTRUCTION IN 1931.

Date.	Great Britain and Ireland.		Abroad.		The World.	
	Steam.	Oil.	Steam.	Oil.	Steam.	Oil.
1st January, 1931	567,715	229,428	774,685	822,234	1,342,400	1,051,662
31st December, 1931	341,781	60,947	758,459	452,478	1,100,240	513,425
Decrease	225,934	168,481	16,226	369,756	242,160	538,237

TABLE III.—MOTORSHIPS UNDER CONSTRUCTION IN GREAT BRITAIN AND IRELAND.

Districts.	Dec., 1931.		Sept., 1931.		Dec., 1930.	
	No.	Tons.	No.	Tons.	No.	Tons.
Aberdeen	—	—	—	—	1	140
Belfast	3	50,130	6	75,957	7	99,300
Clyde :—						
Glasgow	1	6,426	4	28,376	11	71,450
Greenock	1	9,000	3	9,810	8	54,450
Dundee	—	—	—	—	4	32,376
Hull	—	—	—	—	1	190
Leith	1	800	1	800	1	1,000
Liverpool	—	—	—	—	2	12,900
Newcastle	3	22,484	6	45,378	21	150,930
Southampton	2	415	—	—	3	1,150
Sunderland	—	—	2	2,400	4	16,080
Elsewhere	2	600	—	—	1	310
Total	13	89,855	22	162,721	64	440,276

TABLE IV.—MOTORSHIPS UNDER CONSTRUCTION ABROAD.

Countries.	Dec., 1931.		Sept., 1931.		Dec., 1930.	
	No.	Tons.	No.	Tons.	No.	Tons.
British Dominions	—	—	3	850	4	1,830
Belgium	3	1,810	3	1,810	2	1,660
China	—	—	1	850	2	730
Denmark	8	47,300	10	67,215	19	103,430
France	6	58,890	6	58,970	4	44,290
Germany	12	101,656	14	109,878	31	176,650
Holland	23	67,431	30	94,981	43	152,713
Italy	6	77,400	7	56,800	14	76,432
Japan	13	28,700	10	26,240	16	86,060
Lithuania	2	950	2	950	2	950
Norway	5	7,760	8	14,165	5	9,920
Portugal	1	355	1	355	—	—
Spain	9	54,989	9	53,689	11	59,155
Sweden	14	89,550	15	113,950	20	135,350
United States	3	3,437	5	13,007	15	43,263
Total	105	540,228	124	613,710	188	892,433

TECHNICAL PROGRESS IN MARINE ENGINEERING DURING 1931.

TECHNICAL PROGRESS AND TRADE DEPRESSION.

Turning to the technical progress made in the industry during 1931, it is almost a truism that, whenever trade depression is severe, technical development flourishes. For the longest period on record, the industry has experienced an acute depression, and in that time marine engineering progress of outstanding importance has been recorded. In fact, the lean times through which we are now passing will go down in marine engineering history as the period when designers and builders took their courage in both hands, cast off old shibboleths, and truly transformed the face of marine engineering progress.

TEN YEARS' PROGRESS.

Glancing back over the last 10 years, one is reminded of the spectacular triumph of the motorship; the introduction and brilliant vindication of the high-pressure, high-temperature, marine steam-turbine installation of the late Sir Charles A. Parsons; the coming of the double-acting marine Diesel engine; the supercharging of Diesel engines; the gradual change over (not yet completed) to airless injection in marine oil engines; the astonishing popularity, in the Bauer-Wach and similar exhaust-steam turbines, of a discarded idea modernized; the success of the "unsuitable" water-tube boiler for mercantile vessels; the progress in this country of the electric drive for marine propulsion; the development of the exhaust turbo-electric system; the advent of motorships approaching and even exceeding the tonnage of the *Mauretania;* the Diesel-driven warship with geared machinery, having welded framing and weighing less per horsepower than many light-weight high-speed oil engines; the immediate success, practically and technically, of the high-speed heavy-oil engine; "one pound of coal per I.H.P. per hour" in the marine steam engine; specific consumptions of little in excess of ½ pound of oil per S.H.P. per hour with geared-turbine plants; the construction of a great Atlantic liner (for the Compagnie Générale Transatlantique) with turbo-electric machinery; and the building of the Cunard Steamship Company's Blue Riband challenger, with geared-turbine machinery of a power believed to be considerably in excess of that of the existing joint holders of the Atlantic record, or of any of the contenders now being constructed.

The foregoing brief summary covers some of the outstanding achievements of this decade. Who would not say that more progress has been crowded into these last 10 years than was recorded in the previous 25, or 50 for that matter? If order books have not been full for a number of years, those directing the industry at least have the satisfaction of feeling that they have made great strides in technical progress and are ready, now that there are indications of better times ahead, with marine machinery of such efficiency as will attract many shipowners to build tonnage who would otherwise have remained satisfied with their existing vessels. Indeed, it is not an overstatement to say that many of these owners will be obliged to build, or re-engine existing ships, when the full economic significance of these developments in marine engineering begins to make itself felt.

NEW STEAM TONNAGE OF STRAIGHTFORWARD TYPE.

This fact turns one's thoughts immediately to those British owners who during the past year have ordered new steam tonnage of straightforward type, to the exclusion of motorships and steam-driven vessels of the more modern class. Low first cost has undoubtedly been the principal consideration leading to the placing of these orders, and, generally speaking, the machinery specifications have been drawn up with a cold disregard for the progress to which we have just made reference. Normal triple-expansion engines using saturated steam, hand-fired coal-burning Scotch boilers designed for working pressures of 180/200 pounds per square inch, and no "frills," have usually been specified for these ships, which have speeds of the 10/11-knot range. In few instances has superheat been adopted, and the same applies to exhaust-steam turbines and other proved efficiency-increasing appliances. The Diesel engine is similarly being ignored by many British owners who are building at the present time. Abroad, on the other hand, owners are continuing to use the heavy-oil engine, and one or two Dutch owners in 1931 removed steam engines from comparatively modern ships and replaced them with Diesel machinery. The motor-driven "tramp" is being built abroad, and its speed is generally higher than that of the modern British steam equivalent to which reference has been made, while its cruising radius is very much greater.

These facts give one food for thought, and the hope may be expressed that owners will not place the consideration of initial cost before everything else. A scientific assessment of the economic efficiency of any type of projected vessel and its propelling machinery can readily be made, and the best proposition for any particular set of conditions arrived at. These investigations usually show that the simplest form of installation is less efficient (using the term in its broadest sense) than a more economical, but more costly, alternative. For short routes, and for one or two special services involving an unusually small proportion of a ship's total time at sea, the simplest arrangement of machinery is often best. In most cases, however, British owners would do well to spend more money (if available) in obtaining a more efficient vessel—one likely to hold her own against foreign craft which show evidence of their owners having availed themselves of the recent technical developments to which we have alluded.

THE "EMPRESS OF BRITAIN."

The year 1931 is, perhaps, chiefly notable for the completion of the Canadian Pacific liner *Empress of Britain*, the largest mercantile vessel built in a British yard since the war. The largest of the *White Empresses* represents modern turbine propulsion at its best, and is a fitting tribute to the ability and courage of Mr. John Johnson, chief superintendent engineer to the Canadian Pacific Steamships, Ltd. In a very short time Mr. Johnson has, in successive steps, improved the performance of the modern geared-turbine installation, until, in the *Empress of Britain*, he has achieved a figure of about 0.57 pound of oil per S.H.P. per hour for all purposes, which is a really remarkable result when it is borne in mind that the best fuel performance on record before the advent of high-pressure, high-temperature installations was slightly better than 0.80 pound per S.H.P. per hour.

A few months before the completion of the *Empress of Britain*, the *Empress of Japan* was placed in service. The results obtained with this vessel were extraordinarily good and seemed difficult to improve upon. The

superior performance of the *Empress of Britain* is therefore all the more remarkable. In both these vessels three-casing, single-reduction geared turbines of the Parsons type are used; high-pressure, high-temperature steam being supplied by Yarrow oil-fired boilers having very large air-heaters.

THE JOHNSON WATER-TUBE BOILER.

In the *Empress of Britain*, the Yarrow boilers are supplemented by one Johnson two-drum, water-tube boiler. This boiler, it will be recalled, is of Mr. Johnson's own design, and aims at giving a high output from a steam generator of moderate dimensions and weight. The manufacture of this boiler has been taken up by Messrs. Clarke, Chapman & Co., Ltd., of Gateshead-on-Tyne, and already a considerable number of important license agreements have been entered into at home and abroad. This boiler was first tried in the *Princess Helene*, a fairly small Canadian Pacific passenger vessel completed some few months before the *Empress of Japan*. The results obtained with the Johnson boilers of this vessel and of the *Empress of Britain* have, it is understood, been very good, and there is no doubt that the boiler will become popular. It seems to be particularly suitable for naval craft and cross-channel vessels, while its claims for large passenger ships are considerable, as were shown in the paper on "The Effect of Modern Machinery on the Design of Large Ships," read by Mr. A. T. Wall and Mr. H. C. Carey before the Institution of Naval Architects at their summer meetings in Paris.

SUPERHEATING BY ELECTRICITY.

One of the most interesting new developments which calls for comment in the present review is the Lindholmen electrical superheating arrangement, first tried in the cargo steamship *Trione*. It will be recalled that the exhaust steam from the 1000-I.H.P. reciprocating engine of this vessel is passed through the turbine of a geared turbo-generator set, the current from which is passed into heater elements contained in a specially-designed intermediate superheater placed between the high-pressure and intermediate-pressure cylinders of the reciprocator. The electrical superheater reduces the condensation losses and appreciably improves the performance of the installation, the thermo-dynamic efficiency of a normal superheated-steam triple-expansion engine (such as that of the *Trione*) being raised from about 51 per cent to over 82 per cent.

The layout of the arrangement is simple, and the design of the superheater is on practical lines. The superheater consists of two rectangular chambers carrying two sets of tubes expanded into tube plates, the tubes and tube plates being withdrawable for cleaning. The tubes contain electric heating elements, which consist of steatite discs bound with resistance wire. In the *Trione* installation, each heater element takes 1 Kw., the total capacity of the installation being 168 Kw. The steam passes through the two superheating units in series, entering the bottom of the first chamber, flowing around the tubes, leaving the casing by way of a "U"-shaped tube, and entering the second casing at the top. After passing around the tubes in this chamber, the steam leaves at the bottom, the temperature having been increased from 75 to 110 degrees F. As in the case of the Bauer-Wach and

similar exhaust-steam turbine installations, a change-over valve is fitted between the reciprocator and turbine, so that the latter may be cut out, if desired.

The Lindholmen regenerative system is a very interesting alternative to the exhaust-steam turbine driving the same line of shafting as does the main engine, and it will be interesting to see whether it is taken up to any considerable extent. Encouraging results have been achieved with the *Trione* installation; and if it were to be associated with a Lentz poppet-valve equipped, high-pressure cylinder and other Lindholmen economy-increasing features, very high economies would undoubtedly be obtained. As it is, the idea is one which deserves the closest investigation by British engineers, who will watch the performance of the *Trione* with great interest during the coming year.

THE BAUER-WACH EXHAUST-STEAM TURBINE.

The installation of Bauer-Wach exhaust-steam turbines made progress during 1931, although the deepening of the general industrial depression to some extent retarded developments, which is also true of other economy-increasing equipment involving considerable capital outlay. The sponsors of the Bauer-Wach system gave attention to its application to steam trawlers, and a very compact little installation for such vessels was seen on the stand of Messrs. William Beardmore & Co., Ltd., at the last Shipping, Engineering and Machinery Exhibition at Olympia.

Various interests were engaged last year in propaganda for Diesel propulsion for trawlers, for which there is an excellent case. This countermove to the protagonists of steam is very interesting and altogether desirable for all concerned.

THE BROWN-BOVERI EXHAUST-STEAM TURBINE.

The Brown-Boveri interests, it may be recalled, have evolved an exhaust-steam turbine with mechanical coupling, astern blading, and live-steam connections to both ends of the turbine. During 1931 two installations were completed and the vessels—the *Blitar* and *Amasis*—are now in service. Unfortunately, economic considerations did not allow comprehensive trials to be carried out with the equipment in place, but comments on the service results are published elsewhere in this issue. It will be seen that, on the whole, the installations have proved satisfactory and trouble-free.

THE PARSONS EXHAUST STEAM TURBINE.

The Parsons exhaust-steam turbine is a simple, well-designed unit, and its performance in the steamship *Kingswood*, owned by the Joseph Constantine Steamship Line, Ltd., of Middlesbrough, has given every satisfaction. Up to the present, this is the only installation afloat. Some interesting comparative service data for the vessel have been made available, and these show that an increase in shaft horsepower of 26 per cent has been obtained in service on the same steam consumption when the Parsons turbine is brought into series. On progressive trials, over 30 per cent increase was obtained, although only 20 per cent was guaranteed. These results are excellent for a first installation, and, in view of the world-wide reputation of the Parsons Marine Steam Turbine Co., Ltd., the future of this new development seems assured once shipping improves.

THE BENSON SUPER-PRESSURE BOILER.

One of the most interesting developments of 1931, so far as steam propulsion is concerned, was the operation in service of the Hamburg-Amerika Line's turbine-driven steamship *Uckermark* with a Benson super-pressure boiler of 3200 pounds per square inch. This is a bold experiment, upon which the Hapag and their chief superintendent engineer, Dr.-Ing. Emil Goos, are to be congratulated. Service results have been made available, and these indicate that the Benson boiler is giving a good account of itself, and justifies proceeding with the experiment.

The vessel has a single-screw geared-turbine arrangement, with a combination of double and single-ended Scotch boilers supplemented by the Benson "booster" boiler. The steam leaves the Benson boiler at a pressure of 3200 pounds per square inch and a temperature of about 750 degrees F. It is passed through a reducing valve, and is superheated to about 825 degrees F. before it reaches the high-pressure turbine at a pressure of about 880 pounds per square inch. The steam actually passes through six turbine casings, although the Benson boiler and the special high-pressure turbines (which are double-geared) are a separate closed system. The piping arrangements allow either the Benson plant or the normal turbines to propel the vessel.

The *Uckermark* is definitely an unfavorable vessel in which to try the Benson boiler afloat, for the installation is of moderate size, and the incorporation of the special high-pressure turbines and Benson boiler has not been carried out so effectively as would have been the case with an installation specifically designed for the Benson boiler. As it is, some really striking results have been achieved with this ingenious boiler, which is, in effect, a simple coil and special pump designed to produce steam at the "critical" pressure of 3200 pounds per square inch. At this pressure, steam is formed without ebullition or "shock," if such a term may be used, and it is perfectly dry and in every way suitable for driving a turbine.

The efficiency of the *Uckermark's* Benson boiler has been established at 90 per cent, which is remarkably high for marine practice. The vessel has also returned the extraordinary specific fuel-oil consumption of 0.46 pound per S.H.P. per hour, which is equal to the results obtained with Diesel machinery when the usual price differential between the two classes of fuel is taken into consideration. One is aware that in some parts of the world the prices of Diesel oil and boiler oil are identical; but in most places Diesel oil is anything from 15 per cent upwards more costly than furnace oil, and in some ports it is over 60 per cent more costly. It is clear, therefore, that if the *Uckermark* is operated alongside Diesel-driven vessels on routes where there is even a moderate difference in the price of the two classes of oil, the economy of the steamship will be as good as that of the motorships. It is not known what is the cost of the Benson boiler plant, special high-pressure turbines, etc., compared with that of the ordinary high-pressure steam equipment, but it is likely to be fairly high for the present.

While the vessel has admittedly not been in service sufficiently long to allow a really useful opinion to be expressed as to the durability and reliability of the super-pressure equipment, we see no reason why it should not prove satisfactory in these respects. It should be borne in mind, also, that, despite the enormous boiler pressure, there is very little danger of accident due to a boiler explosion. It has been shown that, when a Benson tube fractures, there is little beyond a hissing noise to indicate that an

"explosion" has occurred. We look forward to seeing a large vessel fitted with a Benson super-pressure plant, when, unquestionably, the Diesel-engine advocates will have to look to their laurels.

A DIESEL ALTERNATIVE?

It is developments such as this which force the pace of progress and put designers on their mettle. The Diesel engine accelerated the appearance of the high-pressure, high-temperature steam plant, and steam now seems likely to make it necessary for the protagonists of the marine heavy-oil engine to make another important forward step. The machinery of the *Deutschland*, Germany's remarkable Diesel-driven "pocket" battleship, may be the straw in the wind. As it exists at present, the M.A.N. light-weight, double-acting, geared Diesel installation is obviously not a competitive proposition commercially, but there is no doubt that it could form the basis of a most attractive and very efficient high-powered liner installation. The fuel consumption of such an installation should be very good; while even at the rating which would be adopted for mercantile work, the exhaust-gas temperature would be sufficiently high to allow waste-heat recovery being practicable on such a scale as would permit all accommodation to be steam-heated, certain of the engine-room auxiliaries, etc., being also steam-driven.

TURBO-ELECTRIC PROPULSION.

We have digressed from steam development, however, in our remarks on steam's new challenge to motor propulsion. One of the most interesting features of the year under discussion was the completion of several important British-built turbo-electric liners, such as the *Strathnaver, Monarch of Bermuda, Rangatira* and the United States-built *President Hoover*. The leading particulars of these vessels are given in the accompanying table, together with similar data for a typical Diesel-propelled ship and a geared-turbine vessel—the *Reina del Pacifico* and the *Empress of Japan*, respectively. These two ships are both very economical, and are truly representative of 1931 practice in their respective spheres. They are, moreover, comparative with the electrically-propelled ships on a horsepower basis.

PULVERIZED COAL-BURNING.

Pulverized coal was a subject which was but little heard of during 1931, the two systems which seemed to make most practical headway being the American Todd and the British Clarke-Chapman systems. The former has been applied to one or two additional vessels since our last annual review was prepared, but no information as to the operation of these or of the earlier vessels so equipped seems to have been published during the past year. The Clarke-Chapman system has been taken up abroad, particularly in Germany and Japan, and the results obtained seem to have been good.

Pulverized fuel appears to have been somewhat under a cloud for the past twelve months or thereabouts, and there is no doubt that the problems which have to be overcome are serious. The production of an efficient short-flame burner has not presented great difficulty, and several are now available for use with marine boilers of the Scotch type. Distribution problems present some of the real stumbling blocks which have given much trouble in the past.

	Strathnaver.	*Monarch of Bermuda.*	*Rangatira.*	*President Hoover*	*Empress of Japan.*	*Reina del Pacifico.*
Number of screws	Two	Four	Two	Two	Two	Four
Gross tonnage	22,547	22,424	6,152	21,936	26,032	17,707
Engines (and electric motors, if any).	Two main impulse-type turbo-alternator sets, each rated at 10,700 kW., 3,000 r.p.m., 3,000 volts, three-phase. Two double-unit synchronous motors, each rated at 14,000 S.H.P., 125 r.p.m., 48-pole, 3,000 volts, three-phase.	Two main turbo-generators, each comprising a steam turbine of 10,000 H.P running at 3,000 r.p.m., and 7,500-kW. alternating current generator supplying current at 3,000 volts. Four propulsion motors.	Two main turbo-alternators, each rated at 5,120 kW., 3,080 r.p.m., 3,150 volts, three-phase. Two double-unit synchronous motors, each rated at 6,500 S.H.P., 220 r.p.m., 3,150 volts, three-phase.	Two main turbo-alternators, each rated at 10,100 kW., 2,660 r.p.m., 4,800 volts, three-phase. Two double-unit synchronous induction motors, each rated at 13,250 S.H.P., 133 r.p.m., 4,800 volts, three-phase.	Parsons three-casing, single-reduction geared turbines, 30,000 S.H.P at 120 r.p.m	Harland-B. & W. trunk-piston, airless-injection, Büchi supercharged, Diesel engines; 22,000 B.H.P at about 145 r.p.m.
Main boilers (if any).....	Four Yarrow oil-fired, double-flow, side-fired, water-tube boilers; 400lb/sq. in.; 725 deg. F. final temperature.	Eight Babcock & Wilcox sectional, oil-fired, water-tube boilers; 400lb/sq. in.; 675 deg. F final temperature.	Six Yarrow five-drum oil-fired boilers, 400lb/sq. in., 775 deg. F final temperature	Twelve Babcock and Wilcox standard sectional oil-fired boilers, 300lb/sq. in, 200 deg. F. superheat.	Six Yarrow single-flow oil-fired water-tube boilers; 425lb/sq. in., 725 deg. F. final temperature.	—
Specific fuel consumption, all purposes, lb/S.H.P./hr.	Not available. "All purposes" daily consumption at 20 knots, 27,000 tons displacement, about 162 tons.	—	—	0·676lb per S.H.P per hour.	0·603lb per S.H.P. per hour	0·375lb per B.H.P. per hour.

Both systems mentioned have good distributing arrangements, and this is one of the reasons why they have enjoyed greater success than have certain other schemes.

There are those who hold that the difficulties associated with powdered coal-firing of the Scotch boiler are so great as to be almost insuperable, having regard to the general design of this boiler and the varieties of coal which the average vessel has to bunker in the course of her lifetime. If success of a general character is to be achieved with pulverized fuel-firing afloat, it will be achieved, it is considered by those holding such views, with the water-tube boiler. With large boiler units having generous furnace volumes and a separate pulverizing mill for each boiler, however, most of the trouble encountered with Scotch-boiler installations should disappear.

MECHANICAL STOKERS.

At the same time, now that there is a strong movement towards the more extensive use of coal by British shipping, it may be urged that the claims of the mechanical stoker should be given more consideration in this country. The Underfeed, Erith-Roe and Taylor types have all proved themselves to be sound propositions for marine work, thanks particularly to the enterprise of Dr.-Ir. W. J. Muller, chief superintendent engineer to the Koninklijke Paketvaart Maatschappij, and Mr. John Johnson, chief superintendent engineer to the Canadian Pacific Steamships, Ltd., and it is somewhat surprising that greater headway has not been made in this country. On shore, the mechanical stoker seems to be outstripping powdered fuel for all but the largest boilers, as maintenance costs are less and the mechanical-grate system is cleaner. It is hoped, therefore, that marine engineers will turn their attention to this proved method of burning coal with high efficiency.

TWO NEW STEAM RECIPROCATORS.

One of the most interesting developments of 1931 was the placing in service of a British-designed marine steam engine with poppet valves to the high-pressure cylinder. The North Eastern Marine Engineering Co., Ltd., were responsible for this innovation, which is a straightforward application of the separate poppet-type inlet and exhaust valves to the high-pressure cylinder of an otherwise normal triple-expansion steam engine. The development allows a high superheat to be carried without anxiety, and reduces valve friction, wire-drawing and initial condensation. It should give a very economical engine without raising the initial cost unduly, and more will undoubtedly be heard of this innovation, especially as the smoke-tube type superheaters made by the North Eastern Marine Engineering Co., Ltd., and other firms are now designed with the tubes entering the combustion chamber for a short distance, so as to increase the amount of superheat. The North-Eastern poppet valve cylinder is a logical development emanating from a company who specialize in the manufacture of such superheaters.

From time to time attention has been drawn in these columns to the comparatively poor steam consumption of the general run of marine auxiliary engines suitable for driving circulating pumps, forced-draught fans, dynamos, etc. Very little progress has been made in this field, notwithstanding the fact that the auxiliary load, expressed in pounds of steam per hour, tends to increase as time passes. The appearance of a new and more efficient auxiliary steam engine is therefore to be welcomed, especially when it is the product of such a famous firm of auxiliary specialists as Messrs. Drysdale

& Co., Ltd. The new Drysdale engine was announced towards the end of last year, and is named the "Thermall" engine. This engine is technically as well as practically good, as the designers have sought to maintain cylinder cut-off independent of the point of compression, accuracy of admission and of exhaust periods has been obtained, and wire-drawing minimized. Separate admission and exhaust slide valves are employed in the interests of thermal economy, and a special form of radial valve gear of great robustness and simplicity is used. The circular crank webs are employed as eccentrics for the valve gear, a large crankcase door allowing easy access to the running gear, which is force-lubricated. The ports to the cylinder are arranged tangentially, so that the steam is given a uni-directional flow from inlet to exhaust, the momentum of the steam carrying any condensed water towards the exhaust port and out of the cylinder; this should make for smooth, quiet running. We have no actual comparative data for ordinary and "Thermall" auxiliary engines, but it is understood that the steam consumption of the latter type, *ceteris paribus*, is about 25 per cent below that of the normal auxiliary engine. This is a development which will meet with general approbation if it proves reliable and efficient in service, as no doubt will be the case.

HIGH-SPEED DIESEL ENGINES.

Turning now to oil-engine development, it is probable that the most interesting technical feature of 1931 was the number of new high-speed Diesel engines on view at the Shipping, Engineering and Machinery Exhibition held at Olympia last September. The types now available are very numerous; and when it is realized that almost every engine has a combustion-chamber design which is different from that of any of its competitors, it will be appreciated that great ingenuity is being brought to bear upon the problems of the high-speed Diesel engine. Equally, this great variety of solutions of the problems of efficient combustion and quiet running serves to show that we are far from finality in this particular direction. All these designs cannot be correct, and we are bold enough to assert that many of them are definitely wrong. The next few years will see a levelling up or a "rationalization" of combustion-chamber design in the high-speed engine field.

The wide difference which exists between the best and the worst of high-speed engine specific fuel consumptions is sufficient indication of the truth of the foregoing assertion. These consumptions are fairly comparable, as they are for engines of approximately the same cylinder power and R.P.M. characteristics, and the cylinder dimensions are also comparable. In fact, not a few firms have adopted the actual bore and stroke figures of a certain successful British engine for new designs they have introduced.

Combustion chambers can be divided into three broad types, *viz.*, the open pattern, the pre-combustion chamber style, and the air-cell form. Of these three types, the last-mentioned has proved the most popular for designs of new engines, a notable exception being the new Tangye high-speed unit, which has an open chamber of pent-roof form with inclined valves and a centrally-placed fuel valve—a well-conceived arrangement which should prove very efficient in practice.

Among the air-cell type engines, one of the most notable is the Ruston-Lister, marketed by the newly-formed Ruston Lister Marine Co., Ltd. This engine has two air cells, spheroidal in shape, and arranged in series. The second one, in which the fuel valve is located, communicates with the cylinder by means of a parallel neck. An ingenious feature of this engine is the

provision of a device for ensuring certain starting, even under the most adverse climatic conditions. A screw-down valve is so arranged that it can be made to blank off the secondary air cell when starting, thereby raising the compression pressure and temperature. Even if the operator forgets to return the valve to the "running" position, no harm is done to the engine, for it is designed with this eventuality in mind. The running of the engine is merely less sweet than normally, and there is no danger of parts fracturing.

The ability to start with certainty at all times cannot be overrated: for if these small high-speed Diesels are to make the progress afloat which their economy and general satisfaction warrant, reliable starting under all conditions is essential. They will probably become popular for lifeboats, emergency generating sets, auxiliary compressor engines, and the like; and for all these duties certain starting, even in the coldest weather and with semi-skilled attendants, is a *sine qua non*. Certain of these small Diesel engines are not always dependable starters "from cold," and this is one of the disadvantages of the pre-combustion chamber engine, apart from its inferior fuel performance. It is not surprising to find, therefore, that the pre-combustion chamber principle received little attention last year from designers evolving new high-speed engines.

The Acro type of air cell is really more closely related to the open combustion chamber than it is to the true air-cell principle as used in the Ruston-Lister and other engines. The Acro system is employed in the Ailsa Craig engine—one of the pioneer high-speed marine oil engines—and it has been adopted in the new Dorman Diesel engine which made its appearance last year. This little engine is of about 11 to 12 B.H.P. per cylinder at 1000 R.P.M., and is being made with reduction gear in various sizes; while it is also being offered as an auxiliary or emergency generator engine. The Dorman is an excellent example of the simple straightforward Diesel engine which is laid out for cheap and easy production, and it is a unit which should become popular afloat for various purposes.

While the majority of high-speed Diesel engines are small light-weight machines running at 1000 to 1200 R.P.M., and in design are reminiscent of road-vehicle practice—indeed, one well-known engine designed for marine work has proved outstandingly successful for road transport—there is undoubted scope for the heavier, more robust engine of greater weight and running at a rather lower speed. The Gleniffer engine is a fine example of this class of unit, and is, in the opinion of many, one of the finest real marine-engine designs in the high-speed oil-engine field. It has made a name for itself as a fishing-boat engine, and has also been used for yacht propulsion and other purposes. In designing an engine along "hefty" lines, Gleniffer Engines, Ltd., undoubtedly pursued a sound policy, for which they have been duly rewarded. Their experience and manufacturing resources are such as to indicate that this type of engine was more likely to succeed and meet with the approbation of users and admirers of Gleniffer products than would one of the lighter, quicker-running type.

A new engine of the same class made its appearance at the Shipping, Engineering and Machinery Exhibition last September. We refer to the Widdop high-speed, four-stroke cycle engine, a design of real engineering merit, in which the designer's aim seems to have been, "How good an engine can I produce from the user's standpoint?" Too often the designer has to take as his text, "How good must I make the engine to sell at such-and-such a figure?" While not condemning the latter policy, which often brings out equally able designing, there is real pleasure for the student of design in examining such arrangements as the Beardmore, Gleniffer and

Widdop. The Widdop engine employs the same form of combustion chamber as does the Gleniffer, *viz.*, the Ruston & Hornsby pattern, with horizontal valves and centrally-placed atomizer. There is evidence of careful attention to the avoidance of cylinder-head cracking, as an expansion joint is provided between the inner and outer walls in the region of the fuel valve. A separate liner is fitted, the cast-iron piston is exceptionally long, and the method of driving and disposing the various auxiliaries is in accordance with the best marine practice. In fact, the engine is a real marine production, and should prove very successful for launches, barges, small tugs, picket boats, pilot cutters, etc. The Widdop engine develops 10 B.H.P. at 800 R.P.M. in a cylinder 5 inches in diameter by 5¾-inch stroke. The compression ratio is 15 to 1, and the engine is a cold-starter.

Mention of the Beardmore high-speed engine recalls to mind the interesting fact that last year Messrs. William Beardmore & Co., Ltd., overhauled their high-speed Diesel engine department, introduced several new models, and organized for production on a fairly large basis. They are now building certain types of engines for stock, and the marine field is receiving their attention, together with other spheres. Generator engines for auxiliary and Diesel-electric propulsion are available, and the engine is, of course, suitable for direct-propulsion purposes with reverse gear. The Beardmore engine is somewhat larger than most of the other high-speed engines, the best-known model being the 8¼ inches by 12 inches and developing about 55 B.H.P. per cylinder at 800 R.P.M. In general design the Beardmore high-speed engine remains the same, although cast-iron cylinder heads will be used for marine work instead of the standard aluminium alloy heads. The latest engines are shorter than the example seen on the Beardmore stand at Olympia, for the fuel-pump unit is in future standard engines to be driven from the rear of the engine and located at the side of the crankcase, instead of at the front end of the engine. This change shortens the engine appreciably, and is a welcome one for marine work.

An interesting new high-speed oil engine suitable for marine work is the Aster, the product of a firm who have hitherto made petrol and petrol-paraffin engines for marine work. The Aster engine develops 9 B.H.P. at 1000 R.P.M., and has a patented form of air-cell combustion chamber with external means for raising the compression pressure (and temperature) at starting. This engine is noteworthy for its simple design, the main crankcase and bedplate being a single casting. Messrs. Aster, Ltd., are also offering a high-speed Diesel engine of really advanced design. This has cylinders of the same dimensions as the simple type, *viz.*, 120 millimeters bore by 150 millimeters stroke, and is of the overhead camshaft type, whereas the other engine utilizes push-rod operated valves. The second type, which presents several interesting features in the arrangement of the valve gear, has a rated output of 10 B.H.P. per cylinder at 1000 R.P.M., and has the same form of combustion chamber and cold-starting device as has the other Aster Diesel engine. Speeds up to 2300 R.P.M. are said to be obtainable with this engine, which is available for marine purposes with reverse gear. While it may be considered that these very high speeds are not required for marine work, there is no reason why speeds of the order of 1500 R.P.M should not be used for emergency-generator and lifeboat engines, provided reliability is not impaired. Such fairly high crankshaft speeds are quite practicable with Diesel engines and good fuel consumptions can be obtained, while the purchaser gains in respect to cost and weight per horsepower. Moreover, many of these quicker-running engines are very smooth and quiet in operation, which is indicative of the progress made recently.

MEDIUM-SPEED OIL ENGINES.

There is a commendable tendency for engine-builders to offer quicker-running marine auxiliary oil engines. This development is all the more interesting because a few years ago there was something of a departure from the auxiliary engine running at 250/300 R.P.M. towards speeds of the order of 170/220 R.P.M. Vibration was the cause of this retrograde step in certain cases; and because slower-running auxiliaries overcame such trouble in certain ships, whereas similar vessels with quicker-running auxiliaries had experienced trouble, not a few owners and engineers jumped to the erroneous conclusion that these "high speeds" were wrong, and slower-running units became fashionable. Happily, this mistaken view has been dispelled by actual demonstration on shipboard; and now such firms as Messrs. W. H. Allen, Sons & Co., Ltd., Atlas Diesel Co., Ltd., Belliss & Morcom, Ltd., Burmeister & Wain, Ltd., Crossley Brothers, Ltd., Harland & Wolff, Ltd., M.A.N. Company, Mirrlees, Bickerton & Day, Ltd., Sulzer Brothers, and others, are selling engines in the 300/600-R.P.M. range and above which are reliable, reasonably quiet, free from vibration, and economical.

During the past year this field has been extended by the entry of three firms who are well known in the stationary-engine world, but who have not hitherto catered for the marine market to any extent. The three firms in question are Messrs. Davey, Paxman & Co., Ltd., Messrs. Tangyes, Ltd., and the National Gas Engine Co., Ltd. The first of these three completed a very interesting main-propulsion installation in 1931 in the two five-cylinder Büchi-supercharged engines they supplied for the Diesel-electric vessel *Lochfyne*. These engines, however, represent a design which Messrs. Davey, Paxman & Co., Ltd., are not likely to sell to any extent in the future, in view of the introduction last year of their new so-called heavy-duty type, which is a great improvement on the design of engine installed in the *Lochfyne* in many ways, as well as being lighter and cheaper. The Paxman heavy-duty engine is familiar to readers of this journal by reason of the fact that it was fully described in the December issue. The power per cylinder is 50 B.H.P. at 600 R.P.M. and 62½ B.H.P. at 750 R.P.M., the bore and stroke being 7 inches and 12 inches, respectively. Compactness, rigidity and accessibility are the keynotes of the design, which is laid out for economical production. It will be recalled that this new engine is produced in two forms—the first a single-camshaft unit with cast-iron framing and bedplate, and the second a quicker-running engine with fabricated welded-steel framing and bedplate, two camshafts, and a crankshaft speed, if desired, up to 750 R.P.M. So far as the combustion-chamber shape, horizontal location of the valves, spring-injection system, etc., are concerned, the two forms are identical.

The production of this welded steel-plate Diesel-engine framing and bedplate is a step forward of the greatest technical interest, and may well be the harbinger of important developments in this direction. If reference is made to the illustrations and description of the fabricated bedplate and framing of the Paxman 750-R.P.M. engine, it will be seen that the design has undoubted ingenuity and technical merit. In fact, many would go further and say that this construction is the most meritorious development adopted by a British oil-engine manufacturer in the past two years. The construction is simple, and, unlike the forms of welded framing evolved on the Continent for special engines of high power and low weight, is devoid of numerous small stiffeners and other parts. In a word, it is definitely a

"production" job. A great deal will be heard of this system of framing, it is believed, and it is only a question of time before other makers adopt it or something similar. The weight saved by the steel construction in the case of the Paxman engine is considerable; and as the design is really accessible and has a low overhauling height, it should have a bright future for marine work.

The new Tangye medium-speed engine is another most interesting unit. Here, again, the compact monobloc totally-enclosed construction which is now so popular—and rightly so—has been adopted. The engine is chiefly interesting in having two exhaust valves and one air-inlet valve—an arrangement which is conducive to high volumetric efficiency, freedom from exhaust-valve trouble, and reliability of cylinder covers in service, even at high ratings. The fuel valve is placed in the center of the head, which is a good feature, and the combustion space is compact and of hemispherical form. The engine employs the through-bolt construction—as does the Paxman heavy-duty engine with *cast-iron* framing—and about 70 B.H.P. per cylinder is produced at 500 R.P.M. Especial care seems to have been given to the question of engine balance for all sizes of engine, and the crankshaft is of 40-ton steel, oil-hardened and tempered, and exceeding the requirements of Lloyd's Register of Shipping in the matter of scantlings. This fact, coupled with its clean design and good accessibility, renders the engine very suitable for marine service, and it is to be hoped that it will be given a trial before long.

As in the case of Messrs. Tangyes, Ltd., the National Gas Engine Co., Ltd., have not hitherto built marine oil engines, although their stationary experience is very wide. Their marine generator engine is based on a successful four-stroke cycle stationary type, and is a well-built unit with monobloc construction and totally-enclosed valve gear on the cylinder heads. The National six-cylinder marine auxiliary engine develops 150 B.H.P. at 600 R.P.M., and has vertical valves, separate cylinder liners, and uncooled cast-iron pistons. Quietness of operation is a good feature of the design, but we should like to see more holding-down bolts than had the example at Olympia. This is an easily remedied fault, and the engine is to be regarded as a welcome addition to the ranks of medium-speed, medium-powered generator engines for ships.

It is interesting to note that all the engines discussed in this review as being new to the marine market are of the four-stroke cycle type. Messrs. Burmeister & Wain, Ltd.—for long the stoutest champions of the four-stroke cycle engine—remain one of the few firms offering a small two-stroke cycle engine with separate scavenging. Last year they completed a number of auxiliary-type, two-stroke-cycle, trunk-piston engines, and these, like their large double-acting, two-stroke cycle units, make use of piston-valve controlled exhaust ports and have a "uniflow" scavenge. Airless injection on the firm's own system is employed, and—another very novel feature—the engine has a special form of camparatively slow-speed scavenging blower, which is neatly mounted on the engine. All the running gear is enclosed, and the engine is a remarkably neat-looking little unit. In not a few instances the maneuvering-air compressor is formed integrally with the engine; while in one notable instance—that of the auxiliary-engined ketch *Atlantis*, built at Copenhagen to American account for oceanographic research work—an engine of this type, with four cylinders, has been used for propelling purposes. Moreover, the engine is provided with a mechanical reverse gear—surely the only case on record of a B. & W. engine of 220 B.H.P. at 330 R.P.M. being used for main propulsion, *and* fitted with mechanical reverse gear!

Messrs. Petters, Ltd., are well known as staunch adherents to the two-stroke cycle of operations for their heavy-oil engines. Last year they added an Atomic-type 24-B.H.P. two-cylinder Diesel engine to their already wide range. This engine is made with reverse gear, and should prove an admirable little unit for barges, wherries and other small craft, as it is of the cold-starting type and has a good performance for this class of engine, besides representing the acme of marine oil-engine simplicity.

The Atlas Diesel Co., Ltd., is another concern which has successfully championed the two-stroke cycle engine for marine work. Their latest medium-powered, moderate-speed range of engines of the trunk-piston, airless-injection type, with attached scavenging pump, has already made a great reputation for itself, and is being used by British and foreign owners for tugs, trawlers, small passenger vessels, etc. The range has lately been extended; and whereas the largest size was formerly about 770 B.H.P., an engine of 1160 B.H.P. is made to this monobloc design. Oil-cooled pistons are now used for the large engines in the range. One of the most interesting marine oil-engine developments of the year concerns this well-known design, which is now being manufactured in this country at the Glasgow works formerly owned by Fiat British Auxiliaries, Ltd., and now controlled by British Auxiliaries, Ltd. This concern is in a position to manufacture main-propelling and auxiliary-type engines, as well as stationary units of the same design. The works at Govan being well equipped, this development is an important one for the Atlas interests, especially as it has been closely followed by the imposition of tariff duties on foreign engines and parts. Other foreign engines and accessories popular in this country will also doubtless soon be made in Great Britain.

LARGE DIESEL ENGINES.

Design in the large marine oil-engine field has now settled down, and several proved types have emerged from the bewildering array of designs of a few years ago. The bulk of the orders are now shared by about half a dozen types, which is altogether desirable. The unusual and the technically or economically inferior types have disappeared. Rationalization of design, in effect, is at present the order of things in the marine oil-engine world.

The most important new large marine oil engine to make its appearance last year was of the Sulzer double-acting, two-stroke cycle type. The first example of this straightforward design was installed in the fast single-screw cargo liner *Tajandoen*, owned by the Stoomvaart Maatschappij "Nederland," the unit developing about 7100 B.H.P. at 106 R.P.M. in eight cylinders. Air injection is used, and the engine drives its own reciprocating scavenging air pump. If reference is made to the description of the Sulzer double-acting engine which appeared in this journal, it will be appreciated that the builders have drawn freely upon their marine oil-engine experience with single-acting, two-stroke cycle engines. As a result, the *Tajandoen's* engine has not given the slightest trouble in service, while the performance has also been satisfactory. Engines of the same general type, it will be recalled, are being constructed in the De Schelde shops for installation in the *Indrapoera*, which is being re-engined in order to increase her service speed.

The first M.A.N engine of the single-acting, two-stroke cycle type to have airless injection of the fuel charge was completed towards the end of last year. This engine is of very simple design, and the adoption of airless injection will undoubtedly meet with general approval. The engines of this new type which have already been built are intended for Russian vessels, and are

each of about 2500 B.H.P. in six cylinders. The scavenging air pump is driven off the crankshaft, and in general design the engine follows the well-established double-acting type similar to the 4000-B.H.P. unit which Messrs. Vickers-Armstrongs, Ltd., built at Barrow-in-Furness in 1931. Incidentally, the British engine has turbo-blower scavenging and lever in place of wheel controls. The fuel-pump design has also been modified, mechanical instead of automatic suction valves being provided.

Last year witnessed, in a technical sense, the crowning achievement of the marine Diesel-engine designer in the completion of the machinery for the German pocket battleship *Deutschland* (originally known as *Ersatz Preussen*), which is in many ways the most remarkable warship to be built since the termination of the Great War. The *Deutschland*, it will be recalled, is propelled by twin screws driven by M.A.N. comparatively fast-running, light-weight geared Diesel engines of the double-acting, two-stroke cycle type. There are eight of these engines, each of 7100 B.H.P. at 450 R.P.M.; and they transmit their power through Vulcan hydraulic couplings and single-reduction helical gearing, the final speed of the two propeller shafts being 250 R.P.M. and the shaft horsepower about 54,000. The engines each have nine cylinders 420 millimeters in diameter by 580 millimeters stroke, and it will be noted that the piston speed is remarkably high for such large engines, being about 1700 feet per minute. As is now well known, electric welding has been freely used in the construction of these remarkable engines, a fabricated steel having actually been used for most of the parts which would normally have been of cast iron or cast steel. The weight of the bare engines is about 17 pounds per B.H.P. and of the complete installation about 48½ pounds per B.H.P., the latter figure including shafting, propellers, etc. An interesting feature of these warship engines, which are generally similar to the standard type in basic design, is the employment of rotary valves in each exhaust trunk adjacent to the cylinder. This prevents loss of air through the exhaust system after the scavenge ports have closed, and has allowed higher mean pressures to be carried than was possible without the valves.

Another outstanding Diesel installation of 1931 is that of the quadruple-screw passenger liner *Reina del Pacifico*, owned by the Pacific Steam Navigation Company, which, it will be recalled, has Harland-B. & W. trunk-piston, airless-injection, Büchi-supercharged engines of 22,000 B.H.P. These engines each have 12 cylinders, and the controls are so arranged that the two wing engines are manipulated from the corresponding inboard set, thus greatly simplifying the control arrangements so far as the operating engineers are concerned. Despite the employment of the trunk-piston principle and the adoption of a comparatively high rating, the engines have been very successful and free from trouble; while the fuel consumption is, we believe, about the best which Messrs. Harland & Wolff, Ltd., have ever obtained with oil engines. The trunk-piston principle is attractive for passenger vessels, both from the standpoint of weight and headroom; although something more advanced than the geared double-acting plant of the *St. Louis*, but not quite so ultra-modern as that of the *Deutschland*, would, perhaps, be even more attractive for liner work than is the *Reina del Pacifico* arrangement.

Another fine quadruple-screw motor liner was completed last year in the Lloyd Triestino luxury ship *Victoria*—the fastest motorship in service. This vessel, in which the representative of this journal was privileged to make part of the maiden voyage, has C.R.A.-Sulzer single-acting engines arranged in two engine-rooms, with the auxiliary sets in a third compart-

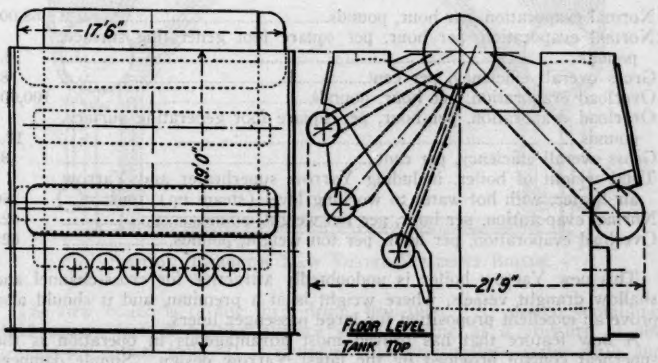
ment. The power of the *Victoria* is considerably less than that of the British vessel just referred to, but in the matter of speed she is superior, having so far averaged about 20½ knots in service.

In drawing this résumé to a close, mention should be made of the twin-screw motorship *Venus*, built for operation on the Bergen-Newcastle passenger service, which is another outstanding vessel of 1931. The power of her engines (5125 B.H.P. each) is high, and the output per cylinder is the highest yet obtained in a mercantile trunk-piston engine. Her Burmeister and Wain engines have their builders' airless-injection gear, as well as the unique B. & W. supercharging system. This makes use of engine-driven blowers, and employs a special form of inlet valve, which allows atmospheric induction in the ordinary way and then admits a supercharging "boost" through a special piston valve incorporated with the main inlet valve. This system has definite advantages, and is proving a success in service on board several vessels, including the Blue Funnel liner *Deucalion*, which recently won the "Wool Derby." The system also appeals to many as having distinct possibilities for auxiliary work.—"The Shipbuilder and Marine Engine-Builder," January, 1932.

DEVELOPMENTS IN YARROW BOILER DESIGN.

Progress Made Within Recent Years and Particulars of the 1932 Type.

The first of the new type Yarrow high-pressure water-tube boilers for the merchant service were introduced in the *Duchess* class vessels of Canadian Pacific Steamships Limited and in the P. & O. liner *Viceroy of India*. These earlier examples are now several years old and it is interesting to note the progress that has since been made as a result of the service experience gained over a considerable period and from further experimental research by Yarrow & Co. Ltd.



The New 1932 Single-Flow, Side-Fired Yarrow Boiler.

Improvements have been made in many directions, and one of the most important advances is the reduction of stokehold area for a given output of steam. This is shown by the following table:—

Yarrow Boiler of	Stokehold Floor Area per 1000 Pounds Steam Evaporated per Hour (From and at 212 Degrees F.)	Corresponding Gross Overall Efficiency
	Square Feet	Per Cent
Early *Duchess* Class	24	85
Strathnever and *Strathaird*	13	88
Latest 1932 Design	9	88

This latest design of Yarrow boiler is illustrated in the outline sketch reproduced herewith. As may be seen from the drawing, the boiler is of the single-flow pattern, with side firing. The Yarrow superheater is placed, as usual, between the upper and lower small drums, while there is a Yarrow multitubular air-heater, although this is not shown in our sketch.

An interesting feature of the design is the raising of the normal evaporation per square foot of generating surface to 8½ pounds as against the 5 to 5½ pounds of earlier designs. It will also be noted that the gross overall efficiency is the very satisfactory one of 88 per cent. The overload evaporation is 12½ pounds per square foot, and at this rating the overall efficiency is only 2 per cent less than the normal value. This latest 1932 design shows a figure of 9 square feet of stokehold floor area per 1000 pounds of steam evaporated per hour from and at 212 degrees F., which constitutes a material improvement on the *Strathnaver* boilers, and a really striking advance on the equivalent figure for the *Duchess* class liners. The following particulars and operation figures will be of interest:—

Normal evaporation, per hour, pounds	68,000
Normal evaporation, per hour, per square foot generating surface, pounds	8.5
Gross overall efficiency, per cent	88
Overload evaporation, per hour, pounds	100,000
Overload evaporation, per hour, per square foot generating surface, pounds	12.5
Gross overall efficiency, per cent	86
Total weight of boiler, including Yarrow superheater and Yarrow air heater, with hot water to working level (steam up), tons	160
Normal evaporation, per hour, per ton weight, pounds	425
Overload evaporation, per hour, per ton weight, pounds	625

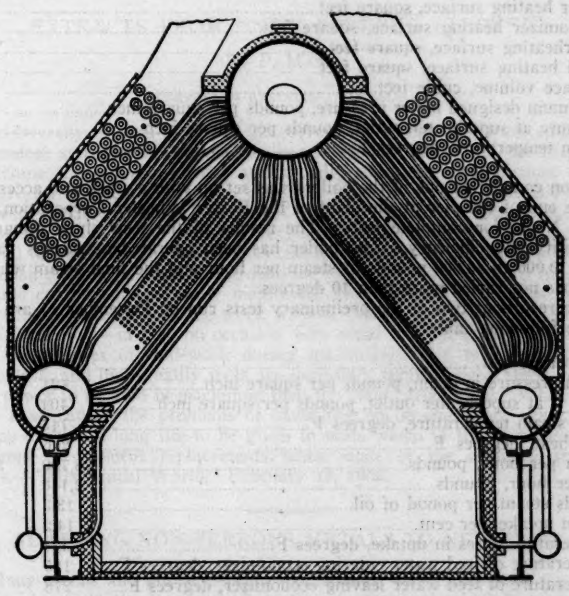
This new Yarrow boiler is undoubtedly attractive for cross-channel and shallow draught vessels, where weight is at a premium, and it should also prove an excellent proposition for large passenger liners.

A new feature that has proved most advantageous in operation is the superheat control provided by the latest Yarrow design. Simple dampers are fitted in the uptakes leading from the boiler and these regulate the

amount of gases passing the superheater and so enable the final steam temperature to be varied as desired over a wide range. Consequently the most suitable degree of superheat is given at all conditions of load and the highest overall efficiency always maintained by the whole machinery installation.—"The Marine Engineer and Motorship Builder," March, 1932.

PRELIMINARY TESTS OF BOILER FOR PANAMA MAIL LINERS.

To provide the increased boiler pressure, higher efficiency and decreased maintenance essential for modern steam propulsion, the Foster Wheeler Corporation, New York, has developed a new steam generator for marine service, the first units of which will be installed in two of the four Panama Mail Liners, the *Santa Paula* and *Santa Elena*, being constructed by Federal Shipbuilding & Dry Dock Company, Kearny, N. J.



SECTION THROUGH NEW FOSTER WHEELER BOILER.

The new boiler is of the three-drum, bent-tube, "A" design containing 4910 square feet of heating surface and operating at a pressure of 400 pounds per square inch. It is equipped with superheaters, water walls and economizers and will burn oil fuel. Four of these boilers will be installed in each of the two vessels.

With the higher boiler pressures now prevailing free water circulation is of prime importance. This led to the choice of the three-drum, bent-tube type of boiler with its large furnace volume and compact arrangement of heating, superheating and economizer surfaces. Instead of supporting the refractory side walls in the furnace with tie bolts the fire rows of tubes have been extended down past the lower drums to headers on a level with the floors of the furnace. These water wall tubes provide efficient cooling of the side walls.

The superheater is of the convection type placed close enough to the floor to give a fairly constant steam temperature regardless of the boiler rating. The economizer sections are installed at the back of the tube banks instead of above the boiler thus permitting the height of the furnace to be increased without projecting the economizer above the fireroom space limits.

The air required for combustion of the oil is blown downward from the top of the rear furnace wall in a double steel casing, thence under the boiler furnace floor and up over the front wall before it reaches the burners.

The boiler characteristics are as follows:

Boiler heating surface, square feet	4910
Economizer heating surface, square feet	3024
Superheating surface, square feet	1955
Total heating surface, square feet	9889
Furnace volume, cubic feet	1120
Maximum designed boiler pressure, pounds per square inch	450
Pressure at superheater outlet, pounds per square inch	400
Steam temperature, degrees F	750

Upon completion one of the boilers was set up with a stack and accessories at the end of the pier at the plant of the Foster Wheeler Corporation, Carteret, N. J., for complete tests. The results of the tests show remarkable flexibility and efficiency. The boiler has been operated at ratings varying from 10,000 to 40,000 pounds of steam per hour and the final steam temperature has not varied more than 10 degrees.

The results of two of the preliminary tests run on December 11 are given in the following table:

Number of test	8	9
Steam pressure in drum, pounds per square inch	421	421
at superheater outlet, pounds per square inch	404	392
Final steam temperature, degrees F	745	755
Superheat, degrees F	296	310
Steam per hour, pounds	28,700	36,250
Oil per hour, pounds	2,146	2,770
Pounds steam per pound of oil	13.3	13.1
CO_2 in uptake, per cent	14.2	14.4
Temperature gases in uptake, degrees F	324	309
Temperature of feed water entering economizer, degrees F	185	153
Temperature of feed water leaving economizer, degrees F	278	271
Moisture in steam, per cent	0.8	0.8
Heat added per pound of water, B.T.U.	1,221	1,262
Heat added per pound of oil, B.T.U.	16,400	16,600
Heat added per hour (in 1000 B.T.U.)	35,200	45,600
Heat in fuel per hour (in 1000 B.T.U.)	40,600	52,200
Heat in fuel per hour per cubic foot combustion space	37,100	47,700
Calorific value of fuel oil, B.T.U. per pound	18,900	18,900

HEAT BALANCE.

	Per cent	Per cent
Overall thermal efficiency	86.5	87.4
Sensible heat loss	5.6	5.2
Latent heat loss	5.4	5.4
Radiation and unaccounted for	2.5	2.0
	100.0	100.0

When installed on the ships the boilers are expected to produce under normal conditions 35,000 pounds of steam per hour. They are guaranteed for an overall efficiency of 85.67 per cent at this rating and the preliminary tests indicate that even better results will be secured. The flue gas temperature leaving the economizer is approximately 300 degrees F.—"Marine Engineering and Shipping Age," January, 1932.

EXTRACTS FROM "NEW FERROUS ALLOYS."

By P. Mabb.

CORROSION-RESISTING METALS.

The diversity of uses to which the corrosion-resisting steels are employed in chemical and general engineering has introduced them into the majority of machine shops. While there are, broadly speaking, three distinct types, they are all of them difficult to machine, particularly the austenitic steels containing 16-20 per cent chromium, 6-10 per cent nickel, with carbon less than 0.2 per cent. These steels are very susceptible to cold-working, and it is essential in machining to ensure that cutting proceeds all the time, otherwise any rubbing action of the tool hardens the material. In drilling, if the pressure and cut be not constantly maintained and the drill allowed to revolve for a time in contact with the work without cutting, it is extremely difficult to finish the hole—in fact, on occasion with small drills it is impossible. To avoid possibilities of cold-work during machining, rigid toolposts, absence of backlash, and high-quality tools are desirable; further, these steels require more power to be used in machining than the plain carbon type. While these steels have caused the problems of machining to become more acute, their use has enabled a long life to be given to units which in the past have been subjected to frequent replacements when made of the ordinary straight metals.—"Mechanical World," February 19, 1932.

ADAPTING NON-FERROUS METALS TO NEW USES.

"Many metals and alloys hitherto mechanically inferior to steel may now be brought up to the standard of steel, as far as mechanical properties are concerned, through the development within the past decade of a process known as age-hardening," Dr. Paul D. Merica, vice-president of the International Nickel Co., said, in the annual lecture of the Institute of Metals Division of the American Institute of Mining and Metallurgical Engineers. "As a result," Dr. Merica continued, "the sphere of usefulness of the non-ferrous metals will be widened. Until comparatively recently, steel has been

practically the only metal that could be hardened and strengthened by heat-treatment. It is for this reason, as well as because of its cheapness, that it has become such an indispensable material of construction.

"The non-ferrous metals—copper, lead, zinc, nickel, aluminum, tin—which are in many respects superior to steel, because of their corrosion resistance and electrical properties, have had limited usefulness in the past because of their mechanical inferiority to steel and because it has not been possible to improve their mechanical properties by heat-treatment. During the last ten years, however, the metallurgist's power to apply heat-treatments usefully in the hardening of non-ferrous metals has been substantially broadened through his understanding of the basic principles of 'age' or 'precipitation' hardening. All the common metals may now be alloyed in such a manner as to render them susceptible to age-hardening. Copper, for example, may be hardened to about 400 Brinell, a value comparable with that of spring steel. Copper-nickel alloys may be heat-treated to show tensile strengths in the neighborhood of 175,000 pounds per square inch, which is also comparable with the strength of heat-treated steel."

Explaining the process of age-hardening, Dr. Merica said: "After certain heat-treatments, these age-hardening alloys harden as they grow older. Sometimes they age, or age-harden, at room temperatures; sometimes they must be aged at higher temperatures. A single day is often adequate to bring about the desired changes in mechanical properties, although slight changes may continue for a month.

"What happens is that there is generated or precipitated throughout the metal or alloy a host of extremely fine, sub-microscopic particles of a hardening constituent. These very fine particles are uniformly distributed throughout the matrix of the metal and harden it.

"The significance of this development in the metal field is that it is now potentially within our power to harden practically any metal or alloy, and thus to put these formerly soft and weak metals and alloys on a comparable footing with steel as regards mechanical properties. The next decade will undoubtedly witness a substantial realization of our promise in this respect."
—"Machinery," April, 1932.

KONEL, A PLATINUM SUBSTITUTE.

In the early radio sets, tubes with platinum filaments were used. With the growth of the radio industry, as well as of other electrical applications of platinum, this metal was found to be rather too scarce and expensive for wide application. Through persistent research, a substitute was found in the form of a new alloy known as Konel, consisting mainly of nickel and cobalt. The new alloy, developed by the Westinghouse Laboratories, has proved itself not only equal, but superior to platinum for the purposes for which it has been developed.

Not only does it have the remarkable resistance to acids and corrosion possessed by platinum, but it has greater strength at red heat and better electrical qualities as a filament. There is one use for platinum, however, for which it cannot serve as a substitute, and that is for jewelry, for it costs not $100 an ounce or more, but only a few dollars a pound. One of the interesting characteristics of Konel is that it retains remarkable strength at high temperatures. In fact, if a steel wire of the same diameter as Konel wire is heated to a red heat, the hot Konel wire will be found to be stronger than the steel wire.—"Journal Franklin Institute," Jan., 32, quoting from "Machinery," Dec., 1931.

PRACTICAL POINTS ON WELDING ALUMINUM.

By Ed. Searles,
Master Mechanic Naval Aircraft Factory, Philadelphia, Pa.

The theory of aluminum welding is well outlined in books on the general subject of welding. The fundamental principles are similar to those governing the welding of other metals in thin sheet form, although a different technique or "feel" is required that can only be acquired by practice. It is the object of this article to give only such facts as have been actually proved in daily practice and to present information of practical value to welders who wish to become skilled in the welding of sheet aluminum.

The equipment used for welding sheet aluminum does not differ in any way from that used for welding light-gage sheet steel, except that a small lead burner's torch is employed. This is necessary owing to the high heat conductivity of aluminum. If a large torch is used, the metal will be burned, and the welds will be large and unsightly. The work will also be distorted by too much heat and the inability to concentrate the flame on the weld.

The small lead burner's torch requires a hose of smaller diameter than that used for the regular full-size welding torch. This necessitates either the use of a reducing valve block or an adapter that is fitted directly to the gages. The oxy-acetylene method may be used, although oxy-hydrogen is used on material 1/16 inch thick or lighter, a clean weld being obtained by the latter method.

TYPE OF WELD FAVORED FOR SHEET ALUMINUM.

Whenever possible, the flange type weld rather than the butt joint is used for sheet aluminum, the flange being burned down by the torch. In this type of weld, a small flange, approximately 1/8 inch high, is turned at right angles to the sheet. These flanges are clamped together tightly and tack-welded. This type of weld requires the addition of very little filler rod, and has the advantage that the flanges help to keep the parts in shape prior to welding. Clamps for holding the flanges together while they are being tack-welded may be made in the shop out of ordinary 1-inch butt hinges, drawn together by means of a bolt welded to one leaf of the hinge, with the end passing through a clearance hole in the other leaf. Ordinary commercially pure soft aluminum wire, usually 1/16 or 1/8 inch in diameter, is used as a filler rod.

SUCCESS DEMANDS THE USE OF THE PROPER FLUX.

It is exceedingly important that a high-grade flux be used, as it is impossible to do high-grade work with an inferior flux. Cost, in this case, is a very poor criterion of quality. Two commercial fluxes that have given satisfactory results in welding aluminum are "Neapolitan" aluminum welding flux, manufactured by the Air Reduction Sales Co., and the aluminum welding flux made by the Aluminum Co. of America.

All aluminum fluxes are hygroscopic and should be purchased only in small sealed glass bottles. Unless the contents are kept airtight, the flux will deteriorate rapidly and poor welds will be the result. The flux is purchased in pulverized form, and a small quantity is mixed with distilled water to the consistency of a very thin paste. It is not advisable to mix more than can be used in one day, and each day the container—usually a porcelain or glass cup—should be washed thoroughly and a fresh supply mixed. The flux is applied to both the work and the filler rod with a small brush similar to an acid swab.

Aluminum should be welded with a neutral flame, as even the slightest excess of oxygen will cause the formation of oxides that will offset the advantages gained by employing the best technique and the highest quality flux. It is far better to use a little too much acetylene than a little too much oxygen. The operator should endeavor to develop as light a touch as possible, and must always be alert for any signs of the metal falling away, due to the application of too much heat. In using a neutral flame, the white inner cone should be kept away from the metal about 1/8 inch and the torch inclined at an angle of about 30 degrees. The so-called forehand or forward method of welding is used in preference to the backhand method. With the forward method, a welder points the torch toward his left and welds from right to left, the flame penetrating the work somewhat in advance of the weld. The torch is usually worked with the acetylene gage indicating a pressure of 15 pounds and the oxygen gage indicating a pressure of 30 pounds.

PREPARATION OF WORK FOR WELDING.

The flanges on the pieces to be welded must be laid out and turned so that they fit neatly and tightly together; otherwise a ragged weld will result. Work having large unsupported areas of light material will often buckle as the result of heat expansion. This can be prevented by producing expansion beads on each side of the weld line. The beads are usually about 1/8 inch in depth and must be reasonably close to the weld or else they will act as stiffeners and defeat the purpose for which they are intended—namely to provide a surplus of metal to absorb the stresses and strains set up by expansion and contraction of the metal due to the heat. If these beads are not provided, the expansion and contraction will cause buckling throughout the entire work.

After applying the beads, the flange seams are placed together and clamped every two or three inches. The hinge type clamps hold the flange plates together tightly and do not slip off. It might be mentioned here that the edges should be wire-brushed before they are clamped together in order to insure clean joints. When the joints are all clamped, the spaces between the clamps should be looked over and any places where the flanges do not lie close together should be squeezed together with flat-nosed pliers.

After this has been done, the work is ready for tack-welding. The welding rod, which is ordinarily pure aluminum wire, usually 1/8 or 1/16 inch in diameter, is slightly heated with the torch for a distance of about 8 inches from the end. The flux is then put on with a small brush, but it is not applied to the seams at this time. The torch is applied to the seam between clamps, and a very small area of the flange is fused, practically no metal being added with the filler rod unless opening of the flanges makes this necessary.

The main use of the filler rod in making the tack weld is to supply the flux for the joint and assist in fusing the flanges when they become molten, in much the same manner that a weld in cast material is puddled. This is accomplished by touching the edges of the flanges lightly with the filler rod at the moment the metal becomes molten. Tack welds are made about every inch, and the clamps are then removed. The seam is next thoroughly cleaned by means of a wire brush. The brushing should be done vigorously, as clean joints at this stage are essential.

The flux is next applied to the seam, coating it thoroughly. The flux is also applied to the wire and the torch played directly on the seam, being

moved back and forth over a space of about 8 or 10 inches until the flux becomes liquid and starts to flow. The seam is then in the proper condition for welding. The remaining procedure is the same as in welding light-gage sheet steel, except that a great deal more care is necessary in order to obtain good fusion. As the flanges are burned down, the molten metal should be constantly agitated with the filler rod to break up the oxide film and thus produce a sound weld.

A great deal of care and practice is necessary to determine the exact moment the metal becomes molten, as there is an extremely short temperature range between the point when it actually becomes molten and the point when it burns and falls away. Determination of the exact moment when the metal becomes plastic is more a matter of instinct, developed by practice, than by color indication, and is one of the things it is practically impossible to describe in writing. Aluminum flux has a decided corrosive action if permitted to remain on aluminum, and should be removed either by immersion or scrubbing with a dilute solution of sulphuric acid (5 per cent) followed by rinsing in hot water.—" Machinery," March, 1932.

A METHOD FOR DETERMINING THE VOLUME CHANGES OCCURRING IN METALS DURING CASTING.

By C. M. Saeger and E. J. Ash.

The present paper reviews methods which have been proposed and used for determining the various types of shrinkage undergone by a cooling metal; defines the three types of shrinkage to be considered—shrinkage of liquid metal, shrinkage during solidification, and shrinkage of the solid metal; and considers in detail two experimental methods, namely, the sand-cast-cone method and the method depending on the determination of the specific volume-temperature relations of the metal.

Reliable shrinkage data were not obtained by use of the sand-cast-cone method, for the following reasons: (a) The temperature of the metal at the instant the mold cavity is completely filled can not be determined, since there is no uniformity of temperature throughout the mass of metal in the mold cavity; (b) a volume change takes place in a portion of the metal within the mold cavity (due to chilling of the metal along the sand surfaces) during the period of pouring the casting, and a certain amount of feeding of the casting has taken place before the mold cavity has filled with metal.

The second and more accurate experimental method used is that which depends upon the construction of the specific volume-temperature curve for the metal under consideration. The specific-volume temperature curve for liquid metal was constructed from data secured by filling a crucible of known volume with the liquid metal at a known temperature. Since the mass of the sample of liquid metal is the same as that of the sample at room temperature, the liquid specific volume can be readily calculated.

Data on the contraction of the solid metal were obtained by direct observations of the change in length with change in temperature of a sand-cast bar of metal. The difference between the specific volume of the solid metal and of the liquid metal at the melting point is the shrinkage resulting during solidification. In the case of an alloy, shrinkage during solidification occurs over a range of temperature. The freezing range of alloys was determined by heating and cooling curves.

Typical experimental data and calculations are given, and shrinkage values for the metal in the liquid state, during solidification, and the solid state were determined for commercial aluminum, red brass (85-5-5-5) and a cast iron. Shrinkage of the liquid metal and shrinkage during solidification have also been determined for an aluminum 8 per cent copper alloy, and for lead, zinc, and tin.—"Bureau of Standards Journal of Research," January, 1932.

(NOTE BY EDITOR: The January, 1932, number of the Journal including this article is for sale by the Superintendent of Documents, Washington, D. C., price 40 cents.)

THE USEFULNESS OF MATHEMATICS TO ENGINEERS.

By P. W. OTT, Assoc. Prof. of Mechanics, the Ohio State University.

The intermediate position of the engineer between the field of pure science and the work of the every-day world has often been set forth. His special work is to dig into the mass of knowledge which has been slowly and laboriously accumulated by mankind and, by selecting and ingeniously combining the necessary facts, to evolve plans whereon he can build machines or structures for the benefit of himself and his fellows.

This task may be anything from the taking of a few simple facts regarding springs and statics and evolving a mouse trap, to the building of a giant battleship, or that magnificent machine at Panama which will lift the battleship over the back of the continent.

Before the engineer may hope for success in such a work, he must be in possession of at least some knowledge of the facts of science and have the ability to compute or estimate the quantitative relations involved in the solution of his problem. These involve not only the elements necessary to make the machine or structure fulfil its purpose, but also the cost, the time necessary for construction, and the economic value of the work.

In short, the engineer must be something of a scientist and mathematician, endowed with foresight, ingenuity and common sense, as well as honesty, courage and vision. Almost any person possesses at least one of these prerequisites. Consequently, any one may engage in engineering work—say, make the mouse trap, or build a fence around his chicken run—and may do so successfully and safely, provided he has sufficient common sense to recognize his limitations.

It is not remarkable that engineers, as a coincidental part of their work, have added materially to the store of knowledge of science and mathematics, for they have not always found the materials and tools ready-made for their hands. In the same way and perhaps to a greater extent, scientists and mathematicians have added to related fields, as well as to their own. Indeed it is difficult to draw a distinct line between any two of these fields.

If a student can recall the order in which the various "branches" of mathematics were presented to him in his formal education, he has them arranged almost in their order of usefulness to him. Perhaps it is fortunate that the student is more gullible at first than later, since it leads to a better mastery of the most fundamental part of mathematics. In his later years of study he becomes skeptical (perhaps rightly) of the utility of certain parts of the subject and tries to acquire that nice degree of mastery which will "get by" the final examination and still not leave his mind cluttered up with useless nonsense!

It is true that the practice of the vast majority of engineers calls for little more than a mastery of arithmetic and geometry and a passing knowledge of trigonometry and algebra. Many of them, in forgetting their higher mathematics, forget also how greatly indebted to it they are for their understanding of science and even for the very origin of the formulas on which they so brilliantly exhibit their skill in arithmetic.

ARITHMETIC THE FOUNDATION.

As for arithmetic, the student must be thoroughly grounded. He must be rapid and accurate, and, above all, must acquire the habit of orderly and systematic computation, so that his operations and results can be easily read and checked. There is usually no course in simple arithmetic given in college, in spite of the fact that the primary and secondary schools do not do their full duty. The student is left to his own devices. He must begin early and keep at it. Addition, subtraction, multiplication, division, involution and evolution, fractions, ratios, and decimals are of every-day usefulness to the student and to the engineer. Learn the short cuts and learn to check. For complete multiplication and division the 9 check-figure method gives an almost sure check. Most engineering calculations involve data of a known degree of accuracy, indicated not by the number of places beyond the decimal point, but by the number of significant figures. Multiplications and divisions with such quantities, when not done by logarithms, should be done by the "bob-tailed" method given by the late Professor Bohannon in his *Plane Trigonometry*.

Let us illustrate this method by multiplying 487.62 by 64.753:

```
487.62
64.753
-------
292572
 19505
  3413
   244
    14
-------
31574.8 or 31575
```

Instead of beginning with the 3, as is usual, begin by multiplying by the 6, bringing down the result, 292572, below the line and draw a line to cancel the last figure of the multiplicand. Then multiply by the second figure, 4, of the multiplier, starting by multiplying the figure just canceled. $4 \times 2 = 8$; this is nearer 10 than 0, so carry 1. $4 \times 6 = 24$, and adding 1 (the carried figure) $= 25$. Place the five below the last figure, 2, of the 292572, carry 2, and proceed, as usual, to multiply by 4 the rest of the multiplicand, canceling the next to last figure, 6, of the multiplicand, when through with the operation. Continue in this manner until all figures of the multiplier have been used. Then add, pointing off the original number of decimal places, in this case 5, less the number of canceled figures, 4. The result is just as accurate as a complete multiplication would have been and about a third of the work has been saved.

The only difficulty here is changing the habit of going from right to left on the multiplier to the reverse. In division, even this trouble is saved, the

only change in habit being to cancel a figure in the divisor whenever tempted to add a zero to the dividend. Multiply each new figure of the quotient by the last canceled figure of the divisor to get the number to carry.

```
                    487.62
                  ---------
                .....
   64.753 | 31574.8
            25901 2
            -------
             5673 6
             5180 2
             ------
              493 4
              453 4
              -----
               40 0
               38 8
               ----
                1 2
                1 3
                ---
```

Note that the first figure, 4, of the quotient is placed directly above the first figure, 2, obtained when the divisor was multiplied by the 4, and that the decimal point of the quotient is put three places (the number of places in the divisor) to the right of the decimal point in the dividend. Get from the library a copy of Bohannon's *Plane Trigonometry* and study carefully the chapter entitled, "Calculation Vices and Devices." Also study the first chapter entitled "Logarithms."

THE NINE CHECK FIGURE.

The fundamental materials of abstract mathematics are *space*, *time*, and *numbers*, and, of these, *numbers* is by far the most slippery. By certain ingenious devices, the practical mathematician manages to fit the thousand and one concepts with which he has to deal into this space-time-number framework, and gets so thoroughly accustomed to the process that he seldom realizes that he is doing it. For example, by the invention of vectors he makes space pinch-hit for a lot of such unrelated ideas as force, magnetic field, electrical potential, and velocity. By the invention of unit quantities, he makes numbers serve a purpose other than counting, for which they were originally invented. If you calculate the cost of 264.7 tons of steel at 47.62 dollars per ton, you are using numbers to calculate a relation having very little to do with *counting*.

There is a large field of study having to do with the properties of numbers and generally called *Theory of Numbers*. One who goes into this field is likely to get a great deal of entertainment out of it, and is certain to be impressed by the ingenuity of those who have worked in it. Although it may not be a very profitable field for the completely pragmatic engineer, since so few of his problems are really numerical in nature, still he does a lot of work with *numbers* as a pinch-hitter, and there are some things he should know about their properties.

Passing over the properties which tell us whether any given number is divisible by 2, 3, 4, 5, 6, 8, or 9 (because almost everybody knows these),

let us consider the peculiar property of the number 9 which enables us easily to check multiplication, division, addition, and subtraction by the "nine check figure system." The nine check figure of any number is the remainder when the number has been divided by nine. In working with it, pay no attention to the decimal point, which is just a little invention to make our old friend *numbers* serve to handle the concept of fractional parts. The propositions about the nine check figure are these: (1) The check figure of the multiplier times the check figure of the multiplicand gives a number whose check figure is the check figure of the product. (2) The check figure of the divisor multiplied by the check figure of the quotient gives a check figure equal to the difference between the check figure of the dividend and that of the remainder. (3) The sum of the check figures of a column of numbers has a check figure equal to the check figure of the sum of the column. (4) The check figure of the subtrahend subtracted from the check figure of the minuend gives the check figure of the remainder. In processes involving subtraction, the desired check figure may come out negative. In this case add nine before using. For example, in checking a problem of division, suppose the check figure of the dividend to be 5 and that of the remainder to be 7. Now subtracting the 7 from the 5 to get the difference referred to in (2) above, we get —2. Adding 9 to the —2, we get 7, which will be the check figure of the product of the check figure of the divisor and that of the quotient.

These checking processes would not be very practical if we actually had to divide every number carefully by nine to find the remainder. Another property of nine enables us to add the separate digits of the number in question together, and then to add the separate digits of the sum obtained, and so on until we get down to something less than nine. Or, more simply still, run the eye along the numbers adding and dropping nine as soon as we accumulate that much or more, gleefully skipping all the nines and all such combinations as a 7 and a 2; a 5 and 4; or a 3, 2, and 4; which happen to strike us as summing nine. It is remarkable how quickly we can acquire the ability to find the check figure almost in a flash.

As an example, let us work the problem about the cost of steel referred to above, that is, multiply 264.7 by 47.62.

264.7	Check figure 1
47.62	Check figure 1
5294	Product 1
15882	
19529	
9388	
11505.014	Check figure 8

There is something wrong with this problem, since the check figures of the multiplier and multiplicand do not, when multiplied, produce a number whose check figure is that of our product. Now to find the trouble: The first line, 5294, is the result of multiplying 264.7 by 2. The check figure of 2 is, of course, 2, and that of 264.7 is 1. The check figure of this line should be 2, and it is. The check figure of the next line should be 6, and is 6. The check figure of the next line should be 7, but it is 8, so there is an error here. The check figure of the last line should be 4, but is actually 1, another error here. Doing these two over, we get 18529 and 10588 for these two lines, with a final sum of 12605.014, whose check figure is 1, as it should be.

It is possible to have a problem which will check by the check figures even though it be wrong, but this will seldom happen. If it does not check by the check figures, it is certainly wrong. The check figure system can be extended to take care of the "bobtailed" multiplication and division by making a little note of the effect on the check figure when a number is cancelled and another carried, but the student with the type of mind which will use this convenience will probably enjoy working it out for himself.

GEOMETRY A BRAIN STRETCHER.

Geometry is a universal preparatory requirement for technical courses. Many of the propositions are of every-day usefulness. The rest, and the logical system of reasoning involved, are necessary for the development of the geometric sense which enables the student or the engineer to translate drawings into ideas and vice versa. It is the foundation for descriptive geometry, probably the most important of the fundamental technical courses, and of all work involving graphical methods—a large field. No student should be satisfied with a half-baked knowledge of "descrip." However, once having gained a real mastery of the subject, he should not be discouraged if he cannot readily recall the solution of particular problems. This ability seems to be retained only by constant practice. The real function of the study is as a brain-stretcher. The ability to "see things in space" never departs.

NEXT TO ARITHMETIC TRIGONOMETRY IS MOST USEFUL.

Trigonometry, next to arithmetic, is the most useful mathematical tool for the engineer. The computing of projections and the solution of triangles must be reduced to permanent mental impressions. Miscellaneous juggling of trigonometric equations may be of less value, but such equations arise with remarkable frequency in engineering computations. To handle them, the student must recognize the fundamental trigonometric identities on sight. If he knows the identities and the rules for solution of triangles "off hand," he can afford to depend on reference to books for the rest.

ALGEBRA IS A HIGH-POWERED MOTOR FOR ARITHMETIC.

The most important thing in algebra is doubtless logarithms, if this branch is to be judged from its practical utility. This is merely a high-powered motor for arithmetic. It speeds up computation, and removes a vast amount of labor, as well as possibility for errors. Students and engineers alike seem to have a tendency to neglect this fine tool, unless it can be said that they use it when they use the slide rule. Perhaps one reason for this neglect is the fear that the trouble of looking up values and interpolating in tables will not be repaid by the saving of labor in computing. There is no excuse for this fear, in the face of modern graphical log tables, which remove all the labor of interpolating as well as most of the chance of reading the wrong value. As for the rest of algebra, the student must be able to solve simultaneous linear equations and quadratics without the necessity of referring to a text. In such solutions, make an orderly arrangement and a numbering of the equations a habit. Indicate what each operation is by such simple statements as "Transpose (3)," "Collect coefficients of x in (4)," "Divide (5) by the coefficient of x^2," etc.

It should be said here that it is useless to be able to juggle equations unless you can originate them, that is, to express relations in mathematical language. Perhaps the ordinary courses in mathematics, from algebra and

trigonometry onward, put too little emphasis on this side of the subject. Lack of ability to do this is certainly the most outstanding mathematical deficiency of upper-classmen in college.

Equations involving higher powers of the variables, and equations transcendental on one side and algebraic on the other occasionally arise in practical work. If the computer knows what he is doing he can make a good guess as to what values will nearly satisfy such equations, and then the method of "cut and try" will lead to a solution with surprisingly little labor. In a cubic, one root by cut and try usually leaves two easily found by the solution of a quadratic.

Let the student be not discouraged if he finds that he has forgotten some of his algebra a year or so after taking, and let him not regret his lost labor in learning it. Skill in juggling equations is invaluable and comes only from practice. It is quite possible for him to be confronted in some actual computation with the problem of solving several dozen simultaneous equations in an equal number of unknowns. This will cause him to compare his formal course in algebra to an introduction to a wall-flower with whom he afterwards has to dance. The strict necessity of conforming to her style makes the analogy quite pat.

"Check and be checked," is the most important phrase applicable to engineering computations. For algebraic work no method beats substituting back in the original equations. Computers are usually paid on a basis of time, rather than the job. This is fortunate.

ANALYT STARTS "CURVE THINKING."

A student taking the usual course in "math" prescribed for engineers will take analytical geometry. This subject is based almost wholly on Descartes' invention of a method of representing equations of one, two, and three variables as curves and surfaces in space. Most people are endowed with what may be called a "geometric mind," and this endowment is deliberately cultivated in an engineering curriculum. Consequently, the ordinary student gets from analytical geometry a new and distinctly gratifying understanding of equations and their meaning and relations, provided, of course, the study "takes" and is not merely something which must be taken. As an engineering tool it is rather limited, but it starts the student thinking in "curves," and thinking in "curves" is so fundamental in the analysis of the vast majority of engineering problems that no labor is ever lost which tends to develop the talent. After taking this course, "curve thinking" soon becomes a fixed habit. Ask any engineering junior what he means by elastic limit of steel, and the chances are he will begin, "It is the point on the curve * * *". From a strict analytical standpoint this is delightfully hazy and wholly incorrect, but he knows what he is talking about and his hearer, with a little esoteric knowledge, knows too. He saves a lot of thinking time, and some words; and from a standpoint of symbolism, he is quite right.

There is a fairly large number of practical engineering problems to which analytical geometry is directly applicable—problems in which space relations are solved analytically, but even without these the subject well merits all the time spent on it in college, and probably more.

CALCULUS IS A VALUABLE TOOL.

Calculus has been made much easier by the concept of curves to represent equations. If the student, on beginning the study of calculus, has a substantial foundation in algebra and analytical geometry, he should have no

difficulty with the calculus, and will get a real "kick" out of the new ideas involved. Otherwise, calculus is very apt to prove a horrible nightmare for him, in which he dreams he is forced to learn the meaning of the meaningless.

As a matter of fact, the engineer who needs calculus is rare, and the one who can use it with facility, still rarer. It is a tool of considerable potential value, which is usually neglected and allowed to rust. Of course the engineer is indebted to it for many of his fundamental formulas. Perhaps it would be wise for the student to resolve to free his mind from the burden of remembering such formulas, and make it a practice to run through the development whenever he needs one. For many of them, the task calls for but a few minutes of time, and the plan would keep the calculus polished and ready for work.

The tool is particularly useful to the engineer in the solution of maximum and minimum problems, a use which involves the easier part of the calculus.

Even if calculus had no use whatsoever in engineering practice, its study would be necessary for an understanding of the work in the last two years of college. It is particularly important that the student really understand the formulas derived by the calculus, even though he can use them with nothing more than arithmetic as an aid.

LEARN TO "JUGGLE" EQUATIONS.

From a poll of engineering teachers made by the Society for the Promotion of Engineering Education, it appears that over 95 per cent of those interested enough to answer the questionnaire believed the above branches of mathematics absolutely fundamental and essential for *all* kinds of engineering. This is a higher rating than was given to any strictly engineering subject. Mechanical drawing got 93.5 per cent and mechanics of materials, 89 per cent. Descriptive geometry got only 70 per cent, which is probably less than it merits on its head-stretching value.

If the student does not have a very thorough knowledge of these branches, he should not attempt anything higher, but should devote his time and energy to a thorough review, working plenty of problems, especially those involving the putting of ideas into mathematical language. If he has a fair knowledge retained from his class work, he will probably profit by using, for his review, different texts from those used in his formal courses.

It should be noted here that for nearly all mathematical studies, no other one thing is more useful than a masterly ability in "juggling" algebraic equations. Any labor spent in acquiring this mastery is likely to be repaid a hundred fold. The student who has burned the midnight oil over algebra will slumber sweetly when the other fellow is having the calculus nightmare. Algebra, to the mathematician, is what practicing scales is to the musician. On the accuracy of the scales depends the harmony of his music, and on the facility of their execution the brilliancy of his technique.

DIFFERENTIAL EQUATIONS.

If the student has a good grasp of the fundamentals, he may with profit pursue some advanced courses. First of these to suggest itself is differential equations (which received a vote of 29 per cent in the poll referred to above). This subject is a superstructure on calculus, having for its main purpose the finding of relations among variables to satisfy any given conditions regarding the variables and their derivations. Expressed in the language of "curves," it attempts to find the curves which satisfy, at every point, any desired relation among the ordinate, abscissa, slope, rate of change of slope, etc., and

then to pick from all such curves the particular ones which pass through given points, are tangent to given lines or meet other required conditions. Needless to say, it is easy to fix up a jumble of relations which can be solved only in theory, but a large number of useful relations *can* be worked out. This subject probably has a greater practical utility to engineers than does calculus. It is an extremely essential tool for researchers and experimenters, and all those engaged in engineering requiring much analytical thinking. In modern electricity it is almost indispensable.

STILL "HIGHER" MATHEMATICS.

With the completion of differential equations, the law of diminishing returns begins to be rather conspicuous in the picture. There are, however, a few more mathematical subjects which can be useful to the engineer, if he stands in a position in his profession which places him near the scientist.

These are, in the order of their probable value:

(1) *Least Squares and Theory of Probability.* This subject cannot be considered a distinct branch of mathematics, but is often given as a college course. It finds its practical application in the adjustment of observations and determination of most probable values and of the probable error of any result of observations and determination of best equations of empirical curves from observed data.

Throughout his technical course, the student is brought into some contact with the theory and an entire lack of knowledge of it is a distinct deficiency in his training.

(2) *Vector Analysis.* This is a distinct branch invented (or, more properly, wrested from the arms of Quaternions) to enable us to handle problems in all kinds of quantities involving magnitude and direction. The student is introduced to it early in his course and makes considerable use of it in an elementary way, by the cumbersome method of resolving on co-ordinate planes to reduce the analysis to algebra. A formal course should clear up his haziness regarding this type of quantities, as well as give him a distinctly new vision and understanding of the ordinary geometrical precepts. Also, this subject has the advantage of being easily grasped, due to its geometric nature. Vector calculus is an extension of the subject of great interest and of much value as a mental stimulant.

(3) *Fourier's Series.* This subject, usually taught in connection with problems in harmonics, develops a method of expressing as one continuous, integrable function any "curve" made up of a number of ordinary algebraic functions. For example, a curve made up of a number of straight lines of different slopes, arcs of circles, ellipses, parabola, and what not, can be expressed in a single equation. The application comes in putting in the limiting conditions for solutions of differential equations, particularly in problems of potential, flow of heat, electricity, stream-line flow, harmonic periods, etc. It is practically worthless to one who has not a good grasp of differential equations, but is a tool of inestimable value in the hands of the skilled mathematician.

(4) *Calculus of Variations.* Perhaps this should be omitted because it is probably not a practical utility to one engineer in a thousand (and also because the writer is profoundly ignorant on this subject himself). The subject develops a method of finding functions to fit given conditions (and particularly maximum and minimum functions) without knowing any relations among the variables or their derivatives, as is the case with differential equations. The method seems to have been invented to solve the problem of the

path of quickest descent. Text books usually begin with an extremely abstruse proof, involving no assumptions whatever, that a straight line is the shortest distance between two points. As an application to mechanics, the writer has succeeded, by this method, in proving that radii remain straight in a round shaft under torsion, and that vertical lines on a beam remain straight when the beam is bent, these being our fundamental assumptions for computing stresses in beams and shafts.

Aside from any possible practical utility, the subject should have great value as a "head-stretcher"; but a head-stretcher is somewhat like a hat-stretcher, which cannot be made to function unless it can be gotten into the hat.

APPLY YOUR MATH AS YOU STUDY IT.

It may be said of all the foregoing courses that if studied aside from their applications, much of their value will be lost. The student should intermingle with them courses in such subjects as thermodynamics, electricity, physics, mechanics, mathematical physics, and physical chemistry. It is only by continual practice on problems already solved that the mathematical method becomes a habit and may be readily applied to new problems. The student should remember that mathematics is a pyramidal structure, and each new part of the superstructure must be laid on the firmest possible foundation. If he is merely building a house of cards, a collapse is nothing more than ludicrous, but if he is making practical use of his creation, a failure may be fatal. The whole process of learning mathematics requires practice and rigid application to problems. The student should hold himself to the task of solving every problem in his text, and some more of his own devising, proving his answers, when no answers are given. No text ever had too many problems. No student who has in view anything beyond a diploma should let "credit" tempt him to take a course for which he is not prepared.

Blind faith in mathematics will not do, at least for the engineer. He must understand its true nature—that it is only a method of reasoning. He must proceed with mathematics in one hand and with common sense in the other, keeping ever a lookout for the reasonableness of his results. Nothing reasonable ever came out of a computation which was not originally put into it in the fundamental assumptions. No calculation can give a result more accurate than the data on which it is based. Mathematical concepts are perfect, the physical conditions to which the engineer applies them never are. All results are in error and it behooves the engineer to know (within limits) how much. His result may be accurate to one-tenth of one part in a million, as in his analysis of the impurities in drinking water, or plus or minus 50 per cent, as in his calculations of the stresses in a structure built to withstand the vagaries of the storms. All formulas are empirical, in that the "rational" ones are derived from empirical ones. None, with the possible exception of Newton's laws, exactly fit the things to which they are applied. When the fiery steed of mathematics is hitched to the block-wheeled cart of utility, every strap of the harness must be properly applied and still the cart will go more slowly and roughly than goes the steed alone.—"The Engineering Experiment Station" (Ohio State University), Circular No. 27, February, 1932.

BOOK REVIEW.

APPLIED GYRODYNAMICS. By Professor Ervin S. Ferry, Professor of Physics in Purdue University. Published in 1932 by John Wiley & Sons, Inc., New York and London. Price, $4.00.

This book has been produced as a text book and reference work for engineers interested in the basic theory and the rapidly increasing applications of the new branch of engineering science which has been called Gyrodynamics.

The author introduces his reader to the subject with a resume of the underlying basic physical principles, and from these develops the mathematical formulae upon which Gyrodynamics itself is based. The book is easy to follow. The difficult subject matter has been developed in sufficient detail with mathematics simple enough to allow a good grasp without the need for additional reference books.

Having based his reader on a good background of principle, the author then describes the Sperry, Brown, Anschutz, Arma, and Florentia navigational gyro-compasses, applying the formulae and principles he has previously presented as he discusses each type. This treatment is followed also in dealing with the gyroscopic stabilizer, airplane monitor control, and the numerous other applications.

The book is recommended to engineers having need in their technical library for a well-developed and complete discussion of this little understood brain-child of pure mathematics.

Commander Roger W. Paine, U. S. N.

ASSOCIATION NOTES.

With regret it is announced that Captain O. L. Cox, U. S. Navy, Secretary-Treasurer of the Society for the year 1931 and re-elected as such for the year 1932, having been detached from duty as Officer-in-Charge of the Design and Construction Division of the Bureau of Engineering, Navy Department, resigned March 1, 1932. On that date he went to duty as Inspector of Machinery at Camden, N. J., with the thanks of the Society for his services of the past fourteen months and with best wishes for his future.

Commander H. F. D. Davis, U. S. N., was appointed by the Council to fill the unexpired portion of the term of Captain Cox.

Many members may be interested in the following announcement:

"CHANGE OF NAME.

"By approval of the Secretary of the Navy dated 3 April, 1932, the name of the Fuel Oil Testing Plant, Navy Yard, Philadelphia, Pa., has been changed, effective 1 July, 1932, to

"U. S. Naval Boiler Laboratory,
Navy Yard, Philadelphia, Pa.

"The serial numbers of tests and reports of the Naval Boiler Laboratory will begin where those of the Fuel Oil Testing Plant ceased. Therefore, confusion will not result if F. O. T. P. Report No. ..... is used when N. B. L. Report No. ..... is meant, or vice versa."

THE ANNUAL BANQUET.

The Annual Banquet of the Society, held in Washington on May 14, 1932, was enjoyed by about 200 members and guests, this number being only slightly less than in other recent years. This

indicates a decided effort on the part of members able to do so to foster this activity of the Society, the only one, it may be pointed out, which serves to bring some of the widely scattered membership into contact with one another.

Rear Admiral F. B. Upham, U. S. N., President of the Society, welcomed the guests and presided. Real Admiral S. S. Robison, U. S. N., Retired, acted as Toastmaster. The speakers and titles of their addresses were:

Admiral W. V. Pratt, U. S. N., "Our Navy."

Rear Admiral S. M. Robinson, U. S. N., "Engineers Ashore and Afloat."

Mr. Carter Field, "The Press and the Navy."

Mr. L. A. Hawkins, "Real Lilliputians of the Universe."

As Mr. Hawkins put it, molecules, atoms and electrons are "The Lilliputians" which, through the electronic tube now in its early stages of development, are destined to exert far-reaching influence on human affairs of the future.

Captain H. S. Howard (C. C.) U. S. N., Captain J. Q. Walton, U. S. C. G., Commander H. F. D. Davis, U. S. N., Commander R. W. Paine, U. S. N., and Mr. H. M. Southgate made the arrangements for the banquet.

MEMBERSHIP.

The following have joined the Society since the publication of the last previous JOURNAL:

NAVAL.

Clark, J. H., Lieut., U. S. N. R., Hotel Fort Selby, Detroit, Mich.

Hartzell, Paul A., Lieutenant, U. S. N.

Lank, R. B., Constructor, U. S. C. G.

Rees, John S., Ensign, U. S. N.

Weil, Louis, Ensign, U. S. N. R., Bennett Hall, Camac Street and Lindley Avenue, Philadelphia, Pa.

CIVIL.

Armstrong, Walter B., Bureau of Engineering, Navy Department, Washington, D. C.

Burtner, Evers, Room 5-428, Mass. Inst. of Technology, Cambridge, Mass.

Carswell, John Stokes, care Lidgerwood Mfg. Co., 775 Lidgerwood Avenue, Elizabeth, N. J.

Clement, Joseph R., Major, U. S. A., Retired, RCA Victor, Inc., 1234 National Press Building, Washington, D. C.

Smith, Ralph Malvin, Defoe Boat and Motor Co., Bay City, Mich.

ASSOCIATE

Kamada, Engineer Lieut. Commander, Imperial Japanese Navy.

Orr, John S., Engineer Commander, Royal British Navy.